STEAM POWER PLANT ENGINEERING

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PREFACE TO THE FIFTH EDITION

A thorough revision of this work has been made to bring it into accord with more recent practice. Advantage is taken of this opportunity to make changes in matter and in arrangement, which it is believed will make it more useful as a text book. While the subject matter has been considerably increased and many new illustrations have been added, the size of the volume has been slightly decreased through the use of narrower page margins and a reduction in the size of the simpler cuts. As in the previous editions, no attempt is made to treat of other than American practice.

G. F. GEBHARDT.

CHICAGO, ILL.,
Jan. 2, 1923

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STEAM POWER PLANT ENGINEERING

CHAPTER I

ELEMENTARY STEAM POWER PLANTS

1. General. — By far the greater part of the mechanical and electrical energy generated for industrial purposes is furnished by the steam power plant. Despite the rapid progress of the internal combustion engine and the intensive development of water power, the steam power plant still leads the field in output and will probably continue to do so for years to come.

The primary object of a power plant is to furnish energy in the desired form, at the point of utilization, at the lowest ultimate cost. The hydro-electric plant requires no fuel in the accepted sense of the term, and the Diesel engine requires less fuel than any other type of prime mover for a given output, so that at first glance it would appear that either of these two would supersede the steam plant, with its extravagant waste of fuel; but the cost of fuel is only one of the many items of expense entering into the ultimate cost of power. Besides, with few exceptions, our large water falls are remote from industrial centers, and the Diesel engine is practically restricted to the use of liquid fuel. Furthermore, the first cost of the hydro-electric plant is usually far above that of a steam-electric plant of equivalent capacity and it is not feasible to transmit the energy beyond a certain limited zone. The Diesel engine has not been considered in this country for large central stations, partly because of the comparatively low capacity of the largest units so far designed (12000 br. hp.). The multiplicity of such units required to meet the demands of our large stations would be objectionable because of space requirements, high first cost, and cost of attendance; but for stations under 5000 kw. rated capacity, the Diesel engine may offer many operating advantages and economies. Because of its exceptionally high thermal efficiency there is a marked tendency toward the increased use of the Diesel engine, not only in the small plants but also in plants of considerable total capacity, consisting of units of relatively large size. The

modern steam-turbine plant is comparatively low in first cost; enormous capacities may be generated in a small space; labor and attendance may be reduced to a minimum; and the plant may be located near the point of power utilization. The low heat efficiency factor of the steam-turbine may be more than offset by these advantages. However, each type of prime mover has a field in which it is superior to the others; but all the factors entering into the problem must be fully considered before an intelligent choice can be made.

Steam power plants may be grouped into two general classes, commonly designated as **isolated stations** and **central stations.** A plant designed to furnish power or heat to a single building is an isolated station, but the term is frequently applied to a plant serving a group of buildings. A plant which distributes power or heat to a number of consumers, more or less distant, is called a central station. When the distances are very great, electrical current of high tension is employed, and is transformed, converted, and distributed at convenient points, through **sub-stations.**

Steam power plants are also classified as condensing and non-condensing, according to the disposition of the exhaust steam. If the exhaust is condensed for the purpose of reducing the back pressure, the plant is said to be operating **condensing.** If the exhaust is discharged at or near atmospheric pressure, the plant is said to be operating **non-condensing,** even if the vapor is ultimately condensed in a feedwater heater or in the coils of a heating system. When the exhaust steam may be used for process work, heating, or other useful purposes, as is frequently the case in various manufacturing establishments and in certain classes of office buildings, hotels, and the like, it is usually more economical to run non-condensing; but where power alone is desired or where only a small part of the exhaust can be utilized, the plant is generally operated condensing. Under certain conditions where varying quantities of exhaust steam are necessary for heating or other purposes, and at the same time considerable power is to be generated, the plant is operated condensing but means are provided for extracting a part of the steam from the engine or turbine in its passage to the condenser.

2. Elementary Non-condensing Plant. — Figure 1 gives a diagrammatic outline of the essential elements of the simplest form of steam power plant. The equipment is complete in every respect and embodies all the accessories necessary for successful operation. Where a small amount of power is desired at intermittent periods, as in hoisting systems, threshing outfits, and traction machinery, the arrangement is substantially as illustrated. The output in these cases seldom exceeds 50 hp., and the time of operation is usually short; therefore the cheapest of appliances are installed, simplicity and low first cost being more important

than economy of fuel. Such a plant has three essential elements: (1) the furnace, (2) the boiler, and (3) the engine. Fuel is fed into the **furnace,** where it is burned. A portion of the heat liberated from the fuel by combustion is absorbed by the water in the **boiler,** converting it into steam under pressure. The steam, being admitted to the cylinder of the **engine,** does work upon the piston and is then exhausted through a suit-

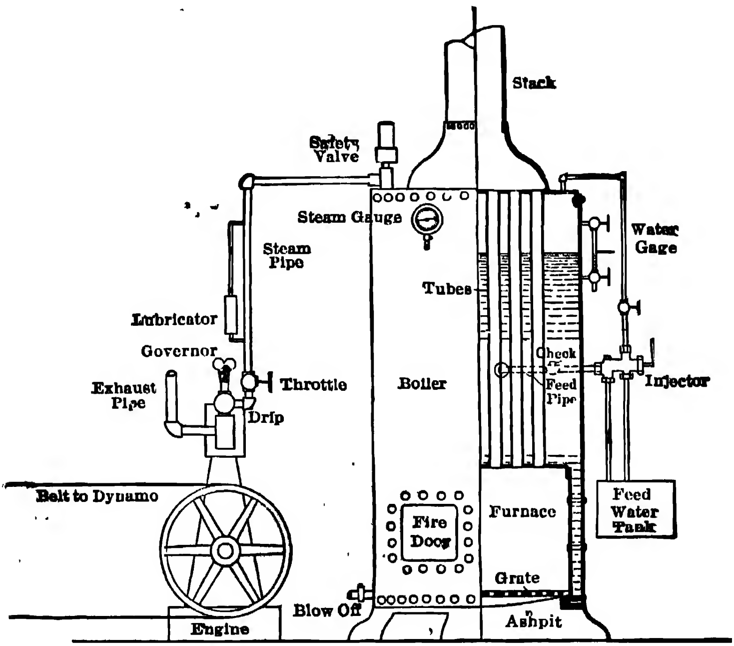

FIG. 1. Elementary Non-condensing Plant.

able pipe to the atmosphere. The process is a continuous one, fuel and water being fed into the furnace and the boiler in proportion to the power demanded.

In such an elementary plant, certain **accessories** are necessary for successful operation. The **grate** for supporting the fuel during combustion consists of a cast-iron grid or of a number of cast-iron bars spaced in such a manner as to permit the passage of air through the fuel from below. The solid waste products fall through or are " sliced " through the grate bars into the **ashpit,** from which they may be removed through the **ash door.** The latter acts also as a means of regulating the supply of air below the grate. Fuel is fed into the furnace through the **fire door,** and when occasion demands, air may be supplied above the bed of fuel by means of this door. The **combustion chamber** is the space between the

bed of fuel and the boiler heating surface, its office being to afford a space for the oxidation of the combustible gases from the solid fuel before they are cooled below ignition temperature by the comparatively cool surfaces of the boiler. The **chimney,** or **stack,** discharges the products of combustion into the atmosphere and serves to create the draft necessary to draw the air through the bed of fuel. Various forced-draft appliances are sometimes used to assist or to entirely replace the chimney. The **heating surface** is that portion of the boiler area which comes into contact with the hot furnace gases, absorbs the heat, and transmits it to the water. In the small plant illustrated in Fig. 1, the major portion of the heating surface is composed of a number of fire tubes below the water line, through which the heated gases pass. The **superheating surface** is that portion of the heating surface which is in contact with the heated gases of combustion on one side and steam on the other. The volume above the water level is called the **steam space.** Water is forced into the boilers either by a **feed pump** or an **injector.** In small plants of the type considered, steam pumps are seldom employed; the injector answers the purpose and is considerably cheaper. A **safety valve** connected to the steam space of the boiler automatically permits steam to escape to the atmosphere if an excessive pressure is reached. The water level is indicated by **try cocks** or by a **gage glass,** the top of which is connected with the steam space and the bottom with the water space. Try cocks are small valves placed in the water column or boiler shell, one at normal water level, one above it, and one below. By opening the valves from time to time, the water level is approximately ascertained. They are ordinarily used in case of accident to the gage glass. **Fusible plugs** are frequently inserted in the boiler shell at the lowest permissible water level. They are composed of an alloy which has a low fusing point and therefore melts when in contact with steam, thus giving warning by the blast of the escaping steam if the water level gets dangerously low. The **blow off** cock is a valve fitted to the lowest part of the boiler, to drain it of water or to discharge the sediment which deposits in the bottom. The steam outlet of a boiler is usually called the **steam nozzle.**

The essential accessories of the simple steam engine include: a **throttle valve** for controlling the supply of steam to the engine; the **governor,** which regulates the speed of the engine by governing the steam supply; the **lubricator,** which is usually of the "sight-feed" class, is attached to the steam pipe, and provides for lubrication of piston and valve. Lubrication of the various bearings is effected by **oil cups** suitably located. **Drips** are placed wherever a water pocket is apt to form, in order that the condensation may be drained. The **apparatus** to be driven by the engine may be **direct connected** to the **crank shaft** or **belted** to the flywheel or **geared.**

In small plants of this type, no attempt is made to utilize the exhaust steam, except in instances where the stack is too short to create the necessary draft, in which case the exhaust may be discharged up the stack. If the draft is produced by convection of the heated gases in the chimney, the fuel is said to be burned under **natural draft**; if the natural draft is assisted by the exhaust steam, the fuel is said to be burned under **mechanical draft.** The power realized from a given weight of fuel is very low and seldom exceeds $2\frac{1}{2}$ per cent of the heat value of the fuel. The distribution of the various losses in a plant of, say, 40 hp. is approximately as follows:

	B.t.u.
Heat value of 1 lb. of coal	11,000
Boiler and furnace losses, 50 per cent	5,500
Heat equivalent of 1 hp.-hr.	2,547
Heat used to develop 1 hp.-hr. (50 lb. steam per hp.-hr., pressure 80 lb. gage, feedwater 67 deg. fahr.	57,500
	Per Cent
Percentage of heat of the steam realized as work, $\frac{2,547}{57,500}$	4.4
Percentage of heat value of the coal realized as work, $\frac{2,547}{57,500 \div 0.50}$	2.2

In Europe, small non-condensing plants are developed to a high degree of efficiency. Through the use of highly superheated steam and specially designed engines and boilers, plants of this type as small as 40 hp. are operated with overall efficiencies of from 10 to 12 per cent, and a 120-hp. unit has been credited with an efficiency of 15 per cent.

The power plant of the modern locomotive is very much like that illustrated in Fig. 1, the main difference lying in the type of boiler and engine. The entire exhaust from the engine is discharged up the stack through a suitable nozzle, since the extreme rate of combustion requires an intense draft. The engine is a highly efficient one compared with that in the illustration, and the performance of the boiler is more economical. In average locomotive practice, about 5 per cent of the heat value of the fuel is converted into mechanical energy at the draw bar.[1] In general, a non-condensing steam plant in which the heat of the exhaust is wasted is very uneconomical of fuel, even under the most favorable conditions, and seldom transforms as much as 7 per cent of the heat value of the fuel into mechanical energy.

Steam turbines are not much in evidence in small non-condensing plants, because their use results in no particular saving in first cost, attendance, maintenance, or space requirements, and the steam consumption per unit output is higher.

[1] Best modern practice gives about 8 per cent as a maximum.

3. Non-condensing Plant — Exhaust Steam Heating. — Figure 2 gives a diagrammatic arrangement of a simple non-condensing plant, differing from Fig. 1 in that the exhaust is used for heating purposes. This shows

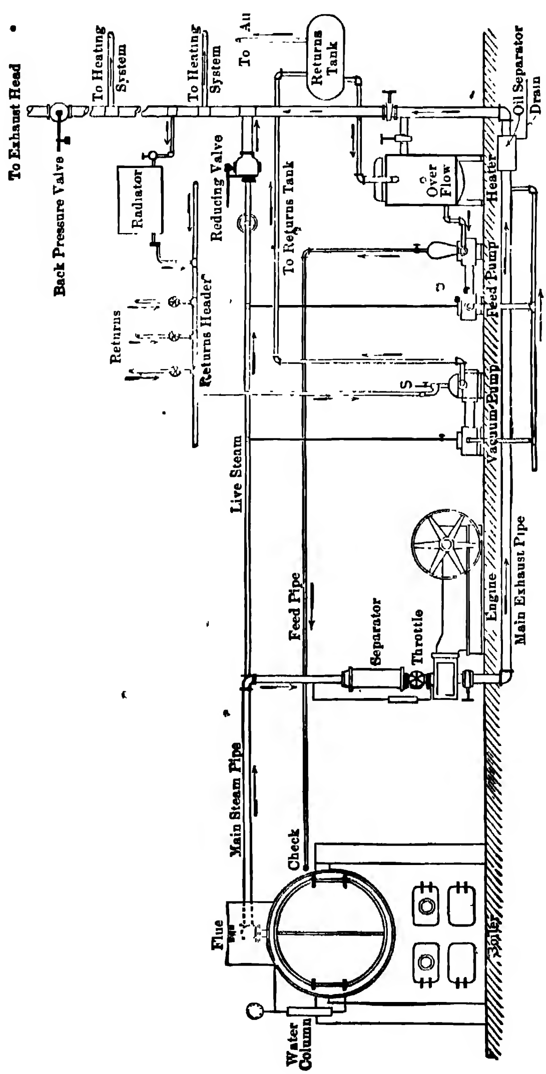

FIG. 2. Elementary Non-condensing Plant with Heating System (Reciprocating Engine).

the essential elements and accessories but omits a number of small valves, by-passes, drains, and the like, for the sake of simplicity. The plant is assumed to be of sufficient size to warrant the installation of efficient appliances. Steam is led from the boiler to the engine by the **steam**

main. The moisture, if any, is removed from the steam before it enters the cylinder by a steam separator. The moisture drained from the separator is either discharged to waste or returned to the boiler. The exhaust steam from the engine is discharged into the exhaust main, where it mingles with the steam exhausted from the steam pumps. Since the exhaust from engines and pumps contains a large portion of the cylinder oil that has been introduced into the live steam for lubricating purposes, it passes through an oil separator before entering the heating system. After leaving the oil separator, the exhaust steam is diverted into two paths, part of it entering the feedwater heater where it condenses and gives up heat to the feedwater, and the remainder flowing to the heating system. During warm weather the engine generally exhausts more steam than is necessary for heating purposes, in which case the surplus steam is automatically discharged to the exhaust head through the back-pressure valve. The back-pressure valve is virtually a large, weighted check valve which remains closed when the pressure in the heating system is below a certain prescribed amount, but which opens automatically when the pressure is greater than this amount. During cold weather it often happens that the engine exhaust is not sufficient to supply the heating system, the radiators condensing the steam more rapidly than it can be supplied. In this case live steam from the boiler is automatically fed into the main heating supply pipe through the reducing valve.

The condensed steam and the entrained air which is always present are automatically discharged from the radiators, by a thermostatic valve, into the returns header. The thermostatic valve is so constructed that when in contact with the comparatively cool water of condensation it remains open and when in contact with steam it closes. The vacuum pump, or vapor pump, exhausts the condensed steam and air from the returns header and discharges them to the returns tank. The small pipe, S, admits cold water to the vacuum pump, and serves to condense the heated vapor and at the same time to supply the necessary makeup water to the system. The returns tank is open to the atmosphere, so that the air discharged from the vacuum pump may escape. From the returns tank the condensed steam gravitates to the feedwater heater where its temperature is raised to practically that of the exhaust steam. The feedwater gravitates to the feed pump and is forced into the boiler. There are several systems of exhaust steam heating in current practice. They differ considerably in details, but, in a broad sense, are similar to the one just described. The more important of these will be described later on.

During the summer months when the heating system is shut down, the plant operates as a simple non-condensing station, and practically all of

the exhaust steam, amounting to perhaps 80 per cent of the heat value of the fuel, is wasted. The total coal consumption, therefore, is charged against the power developed. During the winter months, however, all, or nearly all, of the exhaust steam may be used for heating purposes, and the power becomes a relatively small percentage of the total fuel energy utilized. The percentage of heat value of the fuel chargeable to power depends upon the size of the plant, the number and character of engines and boilers, and the conditions of operation. It ranges anywhere from 50 to 100 per cent for the summer months, and may run as low as 6 per cent for the winter months. This is on the assumption, of course, that the engine is debited only with the difference between the coal necessary to produce the heat entering the cylinder and that utilized in the heating system.

Steam turbines are frequently installed in place of piston engines. The general arrangement of the plant is the same with either type of prime mover, except that no oil separator is necessary for the turbine plant.

4. Elementary Condensing Plant. — Under ordinary favorable conditions, a non-condensing plant cannot be expected to realize more than 7 per cent of the heat value of the fuel as power. In large condensing power stations, the demand for exhaust steam is usually limited to the heating of the feedwater, and as only 12 or 15 per cent can be utilized in this manner, the greater portion of the heat in the exhaust is lost. Non-condensing engines using saturated steam require from 20 to 60 lb. of steam per hour for each hp. developed. On the other hand, in condensing engines, the steam consumption may be reduced to as low as 10 pounds per hp.-hr. The saving of fuel is at once apparent.

Figure 3 gives a diagrammatic arrangement of a simple coal-burning, piston-engine condensing plant in which the back pressure on the engine is reduced by condensing the exhaust steam. A different type of boiler from that in Fig. 1 or Fig. 2 has been selected, for the purpose of bringing out a few of the characteristic elements. The products of combustion, instead of passing directly through **fire tubes** to the stack, as in Fig. 1, are deflected back and forth across a number of **water tubes,** by the **bridgewall** and a series of **baffles.** After imparting the greater part of their heat to the heating surface, the products of combustion escape to the chimney through the **breeching,** or **flue.** The rate of flow is regulated by a damper placed in the breeching, as indicated.

The steam generated in the boiler is led to the engine through the **main header.** The steam is exhausted into a **condenser,** in which its latent heat is absorbed by **injection** or cooling water. The process condenses the steam and creates a partial vacuum. The condensed steam, injection water, and the air which is invariably present are withdrawn

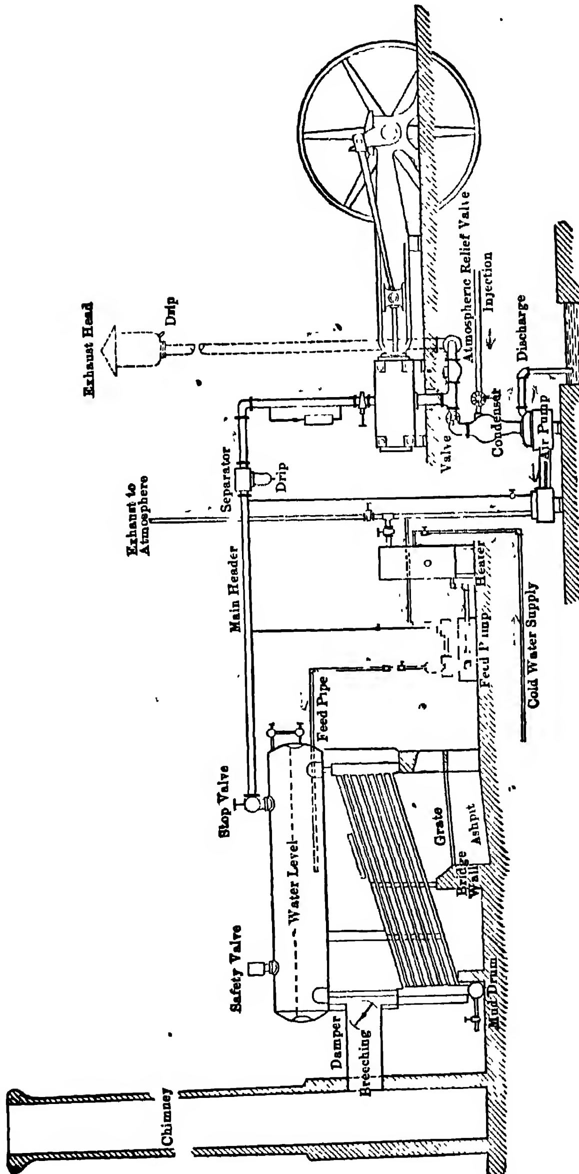

FIG. 3. Elementary Condensing Plant (Reciprocating Engine).

by an **air pump** and discharged to the **hotwell.** In case the vacuum should fail, as by stoppage of the air pump, the exhaust steam would be automatically discharged to the **exhaust head** by the **atmospheric relief valve,** and the engine would operate non-condensing. The atmospheric relief valve is a large check valve which is held closed by atmospheric pressure as long as there is a vacuum in the condenser. When the vacuum fails, the pressure of the exhaust becomes greater than that of the atmosphere and the valve opens.

The feedwater may be taken from the hotwell or from any other source of supply, and forced into the **heater.** In this particular case, it is taken from a cold supply, and upon entering the heater is heated by the exhaust steam from the **air** and **feed pumps.** From the heater it gravitates to the feed pump and is forced into the boiler. Various other combinations of heaters, pumps, and condensers are necessary in many cases, depending upon the conditions of operation. Feed pumps, air pumps, and in fact all small engines used in connection with a steam power plant are usually called **auxiliaries.**

A well-designed station similar to the one illustrated in Fig. 3, when operating under favorable load conditions, is capable of converting about 10 per cent of the heat value of the fuel into electrical energy. The heat distribution under average conditions is approximately as follows:

BOILER LOSSES

	Per Cent
Loss due to fuel falling through the grate	4
Loss due to incomplete combustion	2
Loss to heat carried away in chimney gases	16
Radiation and other losses	8
Total	30

	B.t.u.
Heat used by engines and auxiliaries (16 lb. of steam per i.hp.-hr., pressure 150 lb abs., feedwater 210 deg. fahr.	16,250
Engine and generator friction, 10 per cent	1,625
Leakage, radiation, etc., 2 per cent	325
Total	18,200
Heat equivalent of one electrical hp.-hr.	2,547 B.t.u.
Percentage of the heat value of the steam converted into electrical energy	14.0 (Approx.)
Percentage of heat value of fuel converted into electrical energy $\frac{2547 \times 0.7}{18,200}$	9.8

In Europe, comparatively small piston-engine plants, operating with initial pressure of 500 lb. per sq. in. absolute, initial temperatures of 800 deg. fahr., and vacua of 1 in. absolute, have shown overall efficiencies, at the most economical load, of 20 per cent; and there are a number of

piston-engine central stations in the United States, operating with moderate pressures and superheat, which have realized overall efficiencies of 16 per cent for a short period of time; but, taking into consideration variation in load and all standby losses, efficiencies over 12 per cent are exceptional. These values refer only to the simple condensing plant without economizers, preheaters or heat-saving appliances other than the customary exhaust steam feedwater heaters.

Figure 4 gives a diagrammatic arrangement of one section of a large bulk coal-burning turbo-alternator central station without equipment for reclaiming waste heat. Each section is, to all intents and purposes, an independent plant. It will be noted that the essential elements are practically the same as in the reciprocating station engine plant, Fig. 3, differing only in size and design.

The power house, coal storage pile, storage and switch tracks, overhead bunkers, and coal and ash conveyors have been omitted for the sake of simplicity, though the fuel supply and distributing system is an important factor in the design and operation of the plant. Assuming the coal bunkers over the boilers to be supplied with fuel, the operation is as follows: Coal descends by gravity to the **stokers,** which, in this particular case, are of the underfeed, sloping fire-bed type. Ash and clinkers are removed by **clinker grinders** located in a pit, and are discharged into the ash hopper. Steam- or motor-driven blowers supply the air required for combustion.

The boilers are much larger individually and fewer in number than in the old-style reciprocating-engine plant, and generate steam at 250 to 350 lb. pressure, superheated to approximately 700 deg. fahr. When operating the turbines at full load, the boilers are driven at 175 to 200 per cent or more of their commercial rating. **Reserve** or **spare boilers** are reduced to a minimum. When a boiler is cut out for repairs, the rest of the battery is operated at from 225 to 350 per cent rating or more, in order to evaporate the required amount of water. Each battery is designed to furnish steam directly to one particular turbine, but by means of a crossover main the steam from any battery of boilers may flow to any turbine.

The **prime movers** are horizontal steam turbines direct connected to alternators. The bearings are oil-cooled, and lubrication is automatically effected by means of a pump connected to the governor shaft. Each generator is normally excited by the **main exciter** mounted on an extension of the generator shaft. The generator field may also be excited from an **independently driven exciter** or from the **station storage battery.** Air, washed and conditioned if necessary, is drawn into the generator by centrifugal fans mounted on the rotor, and absorbs the electrical heat

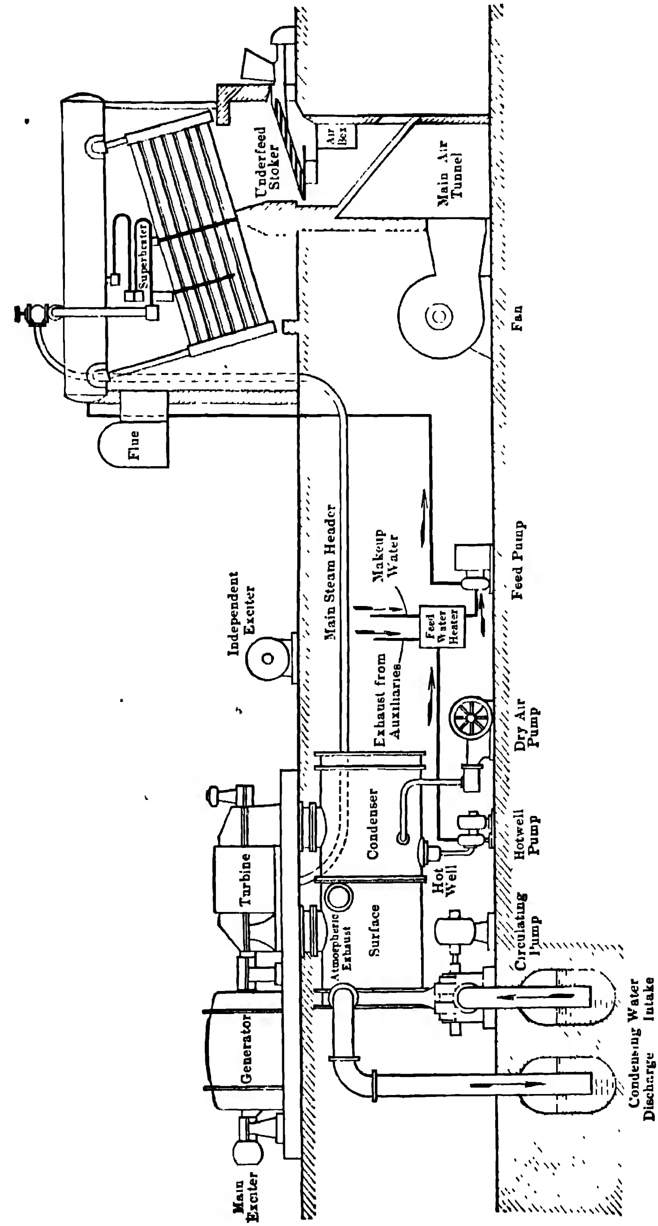

FIG. 4. Elementary Turbo-alternator Station.

losses. The efficiency of the generator is very high (96 per cent), and yet, because of the great amount of energy transformed in the generator, this 4 per cent loss represents a large amount of heat and forced ventilation is necessary to prevent overheating.

The condenser is ordinarily of the surface-condensing type and is attached directly below the low-pressure end of the turbine. A much higher vacuum is maintained in the condensers than in reciprocating-engine practice, since the turbine gives its best efficiency at low back pressures. Condensing water is circulated through the tubes of the condenser by motor-driven or steam-driven centrifugal pumps, and the condensed steam or condensate collected in the hotwells is withdrawn by a turbine-driven or motor-driven hotwell pump. Air and non-condensable vapors are removed by a **dry air pump,** steam or electrically driven. Rotary air pumps, turbo-air pumps and steam ejectors are also used for this purpose. The **hotwell pump** discharges the **condensate** into a feedwater heater, which receives the steam exhausted from the steam-driven auxiliaries. The steam-turbine or motor-driven centrifugal boiler feed pump takes its supply from the feedwater heater and delivers it to the boiler.

A station similar to the one illustrated in Fig. 4, equipped with 20,000-kw. units, is capable of converting over 18 per cent of the heat value of the fuel into electrical energy when operating at its most economical load. Under commercial conditions of operation, with its attendant standby losses, the average overall efficiency ranges from 12 to 16 per cent.

In industrial plants where steam is used for heating or manufacturing purposes, or for both, and where the proportionate demand for low-pressure steam and power would make a straight non-condensing turbine uneconomical, it is common practice to install a **bleeder turbine.** This design is a form of condensing turbine from which steam may be extracted at the desired pressure, either automatically or by manual control. It may be designed for partial or 100 per cent extraction, depending upon the low-pressure steam and the electrical load requirements. In the large, modern power house using electrically driven auxiliaries, the turbines are of the bleeder type, steam being extracted for feedwater heating from as many as four different pressure stages. In certain classes of manufacturing plants where there is an excess of exhaust steam from various steam-using devices, the **mixed-pressure turbine** has been found to give good results. As the name implies, mixed-pressure turbines are designed to run on both high-pressure and low-pressure steam at the same time, using all the low-pressure steam available and sufficient supplementary high-pressure steam to carry the load. When there is sufficient low-pressure steam to carry the load, no high-pressure steam is used; and *vice*

versa, where there is no low-pressure steam available, the turbine functions as a high-pressure machine. **Low-pressure turbines** are operated by exhaust steam only and are installed where there is an ample supply of low-pressure steam to carry the loads at all times. In a number of large central stations, current for the electrically driven auxiliaries and other house service is furnished by a **house turbine.** The exhaust from this turbine (at about 1 lb. gage pressure) is used to heat the feedwater for the entire plant.

The house turbine may be used (1) in conjunction with other steam-driven auxiliaries, (2) when all the auxiliaries are driven by the house turbine with an emergency supply available from the main units, and (3) where current for part of the auxiliary power is furnished by the main units. In the latter case, part of the steam for feed heating is bled from the main unit.

5. Condensing Plant with Equipment for Reclaiming "Waste Heat." — When fuel is costly, it becomes necessary, for the sake of economy, to reduce the heat wastes as much as possible. The chimney gases, which in average practice are discharged at a temperature between 450 and 550 deg. fahr., represent a loss of 20 to 30 per cent of the total value of the fuel. If part of the heat could be reclaimed without impairing the draft, the gain would be directly proportional to the reduction in temperature of the gases. Again, in some types of condensers, all of the steam exhausted by the engine is condensed by the circulating water and discharged to waste. If provision could be made for utilizing part of the exhaust steam for feedwater heating, the efficiency of the plant could be correspondingly increased. In many cases the cost of installing such heat-saving devices would more than offset the gain effected, but occasions arise where they give marked economy.

Figure 5 gives a diagrammatic arrangement of a coal-burning, piston-engine, condensing plant in which a number of devices for reclaiming "waste heat" are installed. The plant is assumed to consist of a number of engines, boilers, and auxiliaries. Coal is automatically transferred from the cars to coal hoppers placed above the boiler, by a suitable conveying system. These hoppers store the coal in sufficient quantities to keep the boiler in continuous operation for some time. From the hoppers the coal is fed intermittently to the stoker by means of a down spout. The stoker feeds the furnace in proportion to the power demanded and automatically rejects the ash and refuse to the ashpit. The refuse is removed from the ashpit when occasion demands, and is transferred to a central ash hopper or dumped directly into cars.

The products of combustion are discharged to the stack through the flue, or breeching. Within the flue is placed a feedwater heater called an

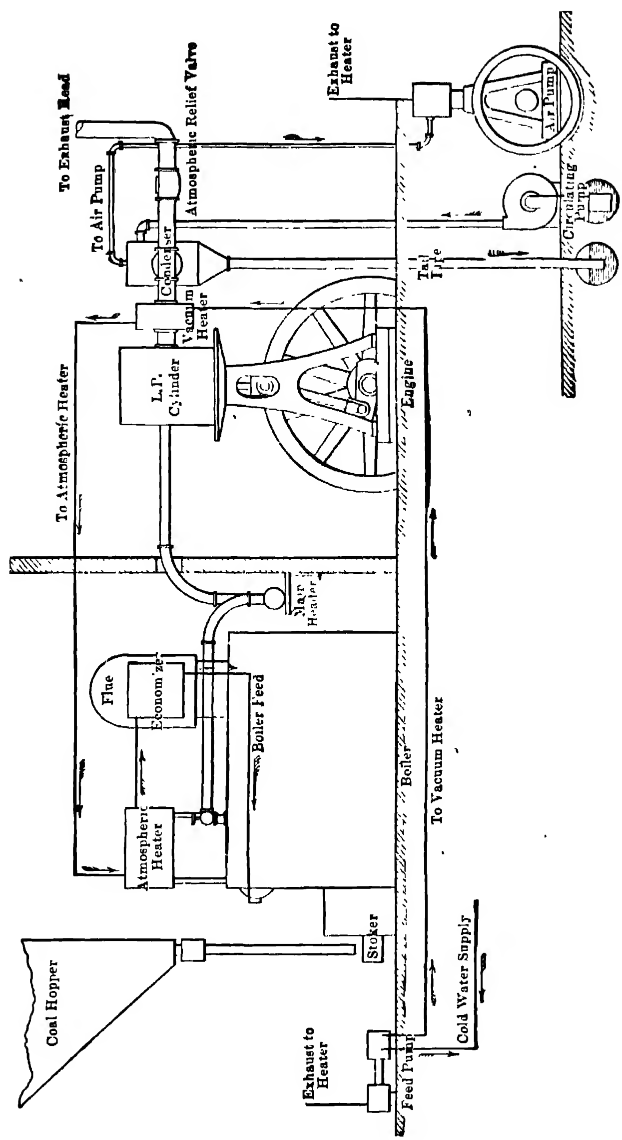

FIG. 5. Elementary Condensing Plant with Various Waste-heat Saving Appliances. (Reciprocating Engine.)

economizer, the function of which is to absorb part of the heat from the gases on their way to the chimney. The heat reclaimed by the economizer varies widely with the conditions of operation and ranges between 5 and 20 per cent. Since the economizer acts as a resistance to the passage of the products of combustion, it is sometimes necessary to increase the draft

either by increasing the height of the chimney, or, as is the usual practice, by using a mechanical draft system.

Part of the heat of the exhaust steam is reclaimed by a **vacuum heater** which is placed in the exhaust line between engine and condenser. For example, if the feedwater has a normal temperature of 60 deg. fahr. and the vacuum in the condenser is 26 in., the vacuum heater will raise the temperature of the feed to, say, 120 deg. fahr., thereby effecting a gain in heat of approximately 6 per cent. If the feed supply is taken from the hotwell, the vacuum heater is without purpose, as the temperature of the hotwell will not be far from 120 deg. fahr.

Referring to the diagram, the path of the steam is as follows: From the boiler it flows through the boiler lead to the main header or equalizing pipe. From the main header it flows through the engine lead to the high-pressure cylinder. The exhaust steam discharges from the low-pressure cylinder through the vacuum heater and into the condenser. Part of the exhaust steam is condensed in the vacuum heater and gives up its latent heat to the feedwater. The remainder is condensed by the circulating water, which is forced into the condenser chamber by the circulating pump. The condensed steam and circulating water gravitate through the **tail pipe** to the hotwell. The air which enters the condenser, either as leakage or as entrainment, is withdrawn by the air pump. The steam exhausted by the feed pump, stoker engine, and other steam-driven auxiliaries is usually discharged into the **atmospheric heater,** which still further heats the feedwater.

Referring to the feedwater, the heat exchange is as follows: The feed pump draws in cold water at a temperature, say, of 60 deg. fahr., and forces it in turn through the vacuum heater, the atmospheric heater, and the economizer, into the boiler. The vacuum heater raises the temperature of the water from 60 deg. fahr. to somewhat less than that of the exhaust steam, or, say, 110 deg. fahr. for average piston-engine practice. The rise in temperature in the atmospheric heater depends upon the amount of auxiliary exhaust available. This ranges anywhere from 8 to 15 per cent of the engine steam requirements for all steam-driven auxiliaries, to 5 per cent or less for combination motor- and steam-driven auxiliaries. In case the auxiliaries are all motor-driven, it is customary to preheat the feedwater either with live steam or with steam bled from the receiver between the high- and low-pressure cylinders, because the introduction of cold water into the economizer causes excessive external corrosion of the latter. When the exhaust from the auxiliaries is in excess of that required to heat the water in the atmospheric heater to 212 deg. fahr., it is usually more economical to drive part of the auxiliaries electrically or with more efficient steam engines. The rise in temperature in the econo-

mizer varies widely with the conditions of operation, but is roughly 0.5 deg. fahr. for each degree drop in flue gas temperature. The heat reclaimed by this series of heaters ranges anywhere from 15 to 25 per cent of the total heat delivered to the engine. If the condenser is of the open, or jet, type and the discharge water is suitable for boiler feed, the vacuum heater is without purpose, because the temperature of the condensate will be practically that of exhaust steam.

The heat distribution in a well-designed piston-engine station, equipped with vacuum heater, atmospheric heater and economizer, and operating under favorable conditions, is substantially as follows:

	Per Cent of Main Engine Steam (I.hp. Basis)		Lb per I.Hp.-Hr. of Main Engines
Steam supplied to engine, initial pressure 165 lb. abs. superheat 100 deg. fahr., vacuum 26-in			12.50
Steam to auxiliaries			
Forced draft fans	1.5		
Induced draft fans	2.0		
Feed pumps	1.5		
Circulating pumps	2.0		
Air pumps	1.0		
Miscellaneous	1.0	9.00	1.125
Steam losses			
Engine and generator friction	7.0		
Leakage, blow-off, etc.	1.5		
Radiation and other heat losses	0.5	9.00	1.125
Total steam required per electrical hp.-hr.			14.75

			B.t.u.
Heat above 60 deg. fahr. required to produce one electrical hp.-hr., 14.75 [1252 − (60 − 32)]			18,054
Heat returned by vacuum heater,	14.75 (110–60)	737	
Heat returned by atmospheric heater,	14.75 (195–110)	1254	
Heat returned by economizer heater,	14.75 (300–195)	1549	3,540
Total heat to be furnished by steam per electrical hp.-hr.			14,514
Per cent of heat of the steam realized as power $100 \frac{2547}{14,514}$			17.5
Boiler, superheater and economizer efficiency, per cent			76
Per cent of heat of fuel required to furnish one electrical hp.-hr. 17.5×0.76			13.3

In Europe, small condensing piston-engine plants, of the locomobile type, operating on substantially the same principles as the one just described, but with very high pressures and superheats, have shown overall efficiency (coal pile to switchboard) of 22 per cent at the most economical

load, and 18 per cent under regular operating conditions; but these results are exceptional. It is quite possible to realize such efficiencies in specially designed piston-engine central stations; but with the present price of fuel, the fixed and operating costs would more than offset the thermal gain. Piston engines are not in evidence in large, modern steam-electric central stations, chiefly on account of their enormous bulk, high first cost, and relatively poor economy. For the smaller central stations, the piston engine, particularly of the uniflow type, is still an active competitor with other sources of power.

Figure 6 gives a diagrammatic layout of a modern steam-turbine plant designed along the general lines of Unit No. 1 of the Waukegan Plant of the Public Service Company of Northern Illinois. In order to avoid confusion in the drawing, the coal- and ash-handling equipment has been partly omitted and the feedwater heating system has been reduced to its simplest elements. Coal is delivered in railroad cars to the station, where it may be unloaded either to storage or to **track hopper.** Although provision has been made in the Waukegan station for the future installation of a **car dumper** and a storage and **reclaiming bridge,** with belt conveyors to carry the coal to and from storage, the present arrangement is to weigh the coal on a **track scale** and dump it into the track hopper, from which it is carried by a single 36-inch belt conveyor to the **breaker** building, 200 ft. distant. At the breaker building, miscellaneous foreign material is separated out and the coal is broken to the correct size for the stokers. From the breaker the coal is delivered to a system of 29-in. belt conveyors, working in a long overhead runway, which feeds it into **cross conveyors.** The latter distribute the coal uniformly into the overhead bins. From these bins the coal gravitates through down spouts to the individual stoker hoppers. Ash is deposited from the end of the grate into a 50-ton ash hopper, from which it is dropped into standard railroad cars.

There are three boilers of the **cross-drum** type for the 25,000 kw. turbo-generator. Each boiler has its own economizer, preheater, and individual induced- and forced-draft fans. Steam is generated at a pressure of 400 lb. gage and superheated to a final temperature of 700 deg. fahr. The stokers are of the **forced-draft, traveling-grate** type, individually driven by a 10-hp. variable-speed alternating-current motor. Air for combustion is supplied by a combination of forced- and induced-draft fans, each boiler having one fan of each type. Each forced-draft fan is driven by a 75-hp. direct connected 440-volt variable-speed motor and has a maximum discharge capacity of 60,000 cu. ft. of standard air per min. against a static head of 2 in. water pressure. Each induced-draft fan is driven by a 150-hp. motor and has a capacity of 92,000 cu. ft. per min. at

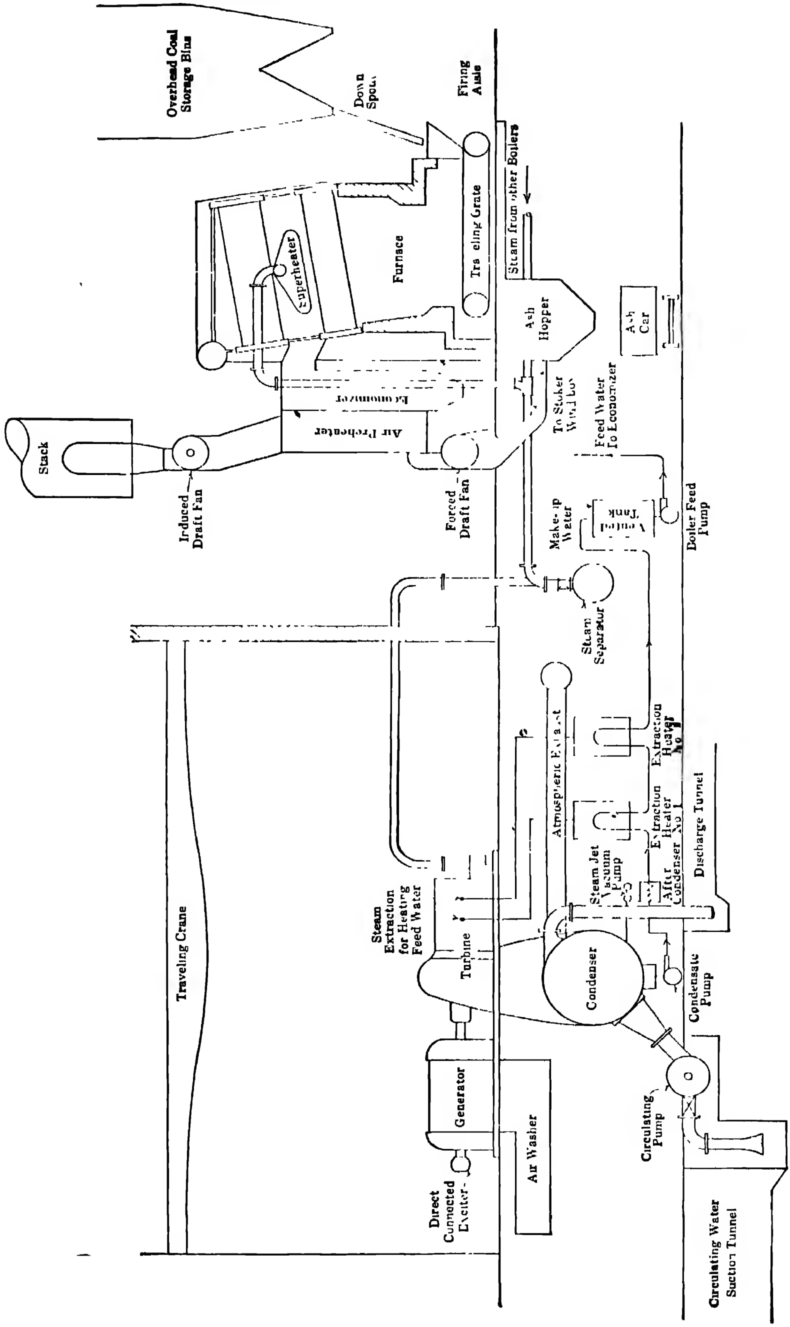

FIG. 6. Elements of a Modern Turbo-alternator Plant with Extraction Heaters, Air Preheaters and Economizers.

350 deg. fahr. temperature. The air is forced through a preheater and its temperature is increased approximately 100 degrees at 200 per cent boiler rating. Control of the speed of fans, as well as that of the stokers and the position of the wind box and uptake dampers, is by manual operation. The induced-draft fans discharge into a single stack, 146 ft. in height above the grate and 15 ft. in diameter.

The prime mover for the unit is a 27,777-kva. single-cylinder, 1800 r.p.m., three-phase, 60-cycle, 12,000-volt Allis-Chalmers turbo-alternator of the bleeder type. Excitation for the generator is supplied by a 200-kw., 250-volt generator direct connected to the main generator shaft. For the purpose of keeping the generator temperature down to the value designated for continuous rated capacity, an open air circulating system, which is capable of handling 75,000 cu. ft. of air per min. is provided. The system is equipped with an air washer.

The condenser is of the surface type and has a cooling surface of 32,000 sq. ft. Cooling water is furnished by a single centrifugal pump having a capacity of 35,000 gal. per min. The pump is driven by a 300-hp. two-speed induction motor having a speed range of 390–435 r.p.m. Condensate is handled by two 2-stage, 600-gal. per min., centrifugal pumps, each driven by a 50-hp. constant-speed induction motor. Air and non-condensable gases are removed from the condenser by two 2-stage evactor steam jets having a capacity of 30 cu. ft. of free air per hour at 1 in. absolute.

Heating of the condensate is as follows: The condensate pump, drawing from the hotwell of the main condenser, forces the condensate first through the **after condenser** of the steam jet air pumps, then through the two **extraction heaters** in series, to a 6000-gal. feedwater storage tank, forming the suction for the boiler feed pumps. The first extraction heater receives steam from the 4-lb. abs. stage of the turbine and the second heater from the 18-lb. abs. stage. The boiler feed pump draws from the storage tank and forces the water through the economizer into the boiler. A third heater, not shown in the illustration, is used for heating the **makeup water.** This heater takes steam from the auxiliaries and from the lower-pressure heating system and is equipped with a steam ejector and condenser for removing non-condensable vapor. A **zeolite** system, having a capacity of 7500 gal. per hr. is installed for reducing the makeup water to zero hardness. The softened water passes through the condenser mentioned above into a 10,000-gal. hot-water reservoir which also receives the high- and low-pressure drips and drains. Either or both extractor heaters may be by-passed, and the condensate from heater No. 1 drains into the condenser hotwell while the condensate from heater No. 2 goes to the hot-water reservoir. The feedwater storage tank also overflows to the reservoir.

There are three 500-gal. per min. boiler feed pumps per turbine unit, two driven by 250 hp., 2275 r.p.m. steam turbines and the other by a 350 hp., 1700 r.p.m. induction motor. The power required to operate all the auxiliary apparatus at full generator capacity is approximately $3\frac{1}{2}$ per cent of the total output.

In some of the very latest large central station installations, boiler pressures of 550 lb. gage and total steam temperature of 750 deg. fahr. are employed, and small boiler units of 1200 lb. are in course of construction in at least two new projects. In nearly all of the latest plants, the station auxiliaries are all motor-driven except certain standby units which are steam-turbine-driven. The **duplex drive** for certain auxiliaries is also in favor. With this system the auxiliary is driven by a motor and steam turbine connected to a common shaft, the motor being used for normal operation and the steam turbine for emergency. The electrical losses are also partially recovered, in some instances, by using the condensate for absorbing the heat from the generator ventilating air. The heat rejected by steam-ejector air pumps, high-pressure gland steam, low-pressure gland water, makeup-water evaporators, drips and bearing oil colors is frequently used for heating the condensate or feedwater. Deficiency in feedwater temperature for the required "heat balance" is also effected by bleeding the main turbine progressively at one to four points. Feedwater economizers for utilizing part of the sensible heat of the flue gases are the rule rather than the exception, though increased feed temperatures from stage bleeding render them less desirable. In case of multi-stage bleeding and high-pressure and temperature steam with stage reheating, which is now being considered for a number of new projects, feedwater economizers are not to be included, and the waste heat from the boiler is to be utilized in preheating the combustion air. Preheating air by bleeding the main turbine unit at one of two stages is a feature in two of the new plants, but no data are available as to savings effected. A combination mercury-steam unit of approximately 4000-kw. capacity is in commercial service at the Dutch Point Station of the Hartford Electric Light Co. In this unit, mercury is vaporized in a special boiler at a pressure of 35-lb. gage and corresponding temperature of 812 deg. fahr., and expanded to a 29-in. vacuum in a mercury turbine. The mercury exhaust, at a temperature of 414 deg. fahr., is condensed, and its latent heat is used for generating steam at about 200-lb. gage pressure. This steam is used for operating the conventional type of steam turbine. Compared with an efficient steam-turbine plant operating at 200-lb. pressure, the mercury-steam combination gives about 50 per cent more electrical output per lb. of fuel. To what extent these refinements may be carried out without offsetting the heat economy, by increased first

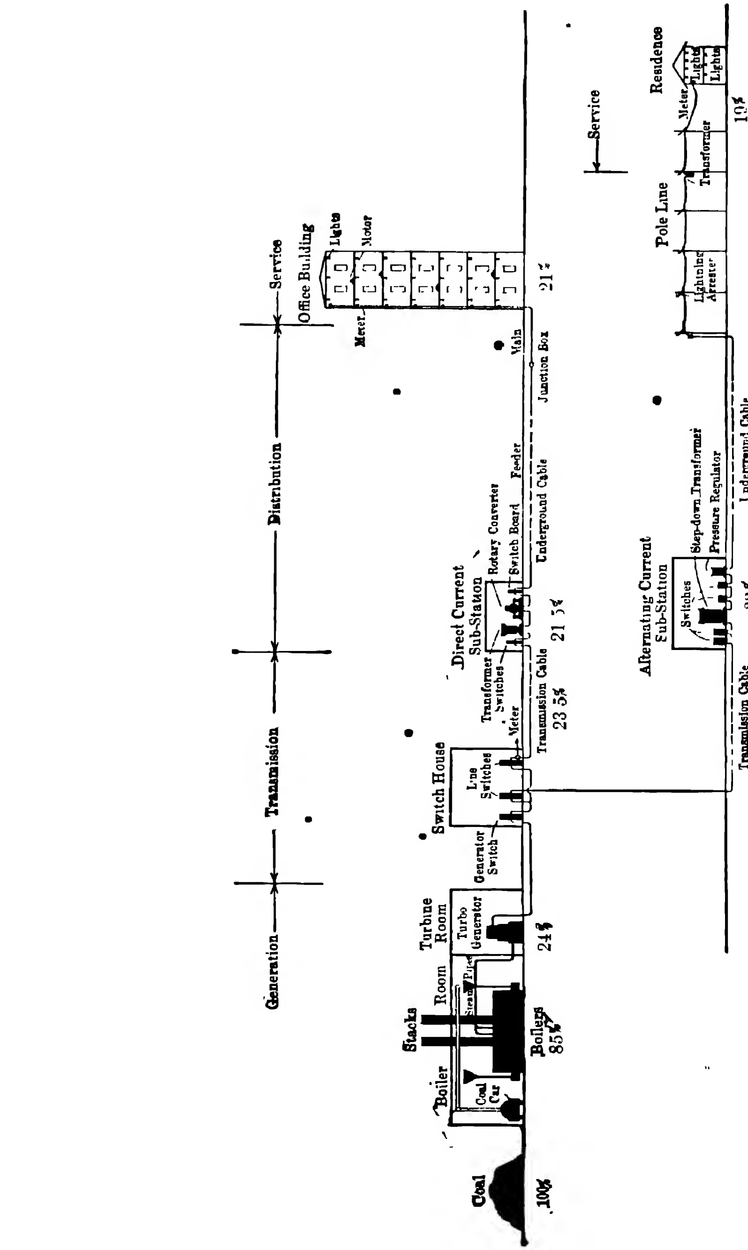

Fig. 7. Approximate Distribution of Energy Losses in a Modern Turbo-alternator Central Station.

cost, maintenance, attendance and interruption to service, can be determined only by careful analysis of all the factors entering into the problem. The use of pulverized fuel, and in certain locations fuel oil, has resulted in reduced **standby losses** and increased overall station economy; but this is only another one of the many factors in the problem of economical power generation. If the predicted results of some of the latest central station projects are realized, a kilowatt-hour may be developed on a heat consumption of 12,500 B.t.u., corresponding to an overall efficiency of 27.4 per cent, or "a pound of coal per kilowatt-hour."

The percentage of the heat value of the fuel realized as energy at the point of consumption is considerably less than the overall efficiency from "coal-pile to switchboard," because of the transmission, distribution and service losses. These losses vary within wide limits, depending upon the size and type of plant, character of equipment, length of transmission lines, and various other influencing factors. Figure 7 illustrates the approximate losses for a large plant such as the Crawford Avenue Station of the Commonwealth Edison Company of Chicago.

6. Superpower Plants. – Any of our ultra-modern turbo-alternator central stations is in effect a superpower plant, but the word "superpower" in this connection is intended to refer to the large central stations comprising a part of a regional system, advocated by Wm. S. Murray, through which all the large load centers within its boundaries will be linked together. This system will include large base-load steam-electric plants at tidewater, on inland waters or in the coal-mining territory, as conditions may warrant, and hydro-electric developments at potential water-power sites. These new base-load plants will be linked together with the more efficient existing plants, by means of heavy trunk transmission lines capable of carrying sufficient power to permit each power plant to be operated in the manner necessary to get the greatest resultant economy for the entire region. This system calls for the ultimate electrification of the railroads and the supplying of electrical energy for all industrial and domestic requirements within the prescribed zone. That large savings will accrue from the regional power system is evidenced by the savings already effected through the extension of our modern central stations. The whole question of superpower, however, despite its great possibilities for good, is not one of engineering but rather of finance, and the reader is referred to the accompanying bibliography for extended study.

The Performance and Cost of the Superpower System: Power, Nov. 8, 1921, p. 725.
Economies to be Expected of Superstation. Power, June 1, 1920, p. 879.
Superpower Transmission: Jour. A.I.E.E., Feb., 1924, p 3.

EXERCISES

1. Make a diagrammatic outline of a simple non-condensing plant, correctly locating all the essential elements entering into its composition. Indicate by means of arrow points the direction of flow of the feedwater and steam.

2. Same instruction as in Problem 1, except that a non-condensing plant with exhaust steam heating system is to be considered.

3. Enumerate the character and extent of the heat losses from "coal-pile to switchboard" in a simple non-condensing piston-engine plant.

4. Beginning with the cold water supply, trace the path of the feedwater and steam through the various essential elements in a condensing plant equipped with a full complement of "heat-saving" appliances.

5. Make a skeleton outline of a modern turbo-alternator plant, correctly locating and designating by name all the essential elements entering into its composition.

CHAPTER II

FUELS

7. General. — The cost of fuel is by far the greatest single item of expense in the production of steam power, and ranges from 40 per cent to 70 per cent of the total operating expense. Furthermore, all fuels are slowly but surely increasing in price, and larger investments for fuel- and labor-saving equipment are justified. In localities where a specific fuel is plentiful, the problem resolves itself merely into a study of the best methods of burning this fuel; but in situations where various kinds of fuel are available, the selection of the one best suited for a given or proposed equipment includes a careful consideration of such items as composition of the fuel, size, cost per unit, heating value, refuse incident to combustion, initial waste products, such as ash and moisture, storage requirements, and transportation facilities.

Where a choice exists, that fuel is selected which develops the required power at the lowest cost, taking into consideration all of the circumstances that may affect its use. Occasionally the disposition of waste products is a factor in the choice, but such instances are uncommon. The boilers and furnaces are designed to suit the fuel selected.

American Fuels, Bacon and Hamor, Mc-Graw Hill Pub. Co.

8. Classification of Fuels. — The fuels most commonly used for steam generation may be divided into three classes, as follows:

1. Solid fuels.
   - *a.* Natural: coal, lignite, wood, and peat.
   - *b.* Prepared: coke, charcoal, pitch, and briquetted fuels.
2. Liquid fuels.
   - *a.* Natural: crude oils.
   - *b.* Prepared: distilled oils, residuum, gas-works tar, gas oil.
3. Gaseous fuels.
   - *a.* Natural: natural gas.
   - *b.* Prepared: blast-furnace gas, coke-oven gas, coal gas, water gas, producer gas, oil gas.

The majority of prepared fuels are by-products of manufacturing processes.

SOLID FUELS

9. General. — Solid fuels are of vegetable origin and exist in a variety of forms, ranging from a comparatively recent cellulose growth to graphitic anthracite coal, which is nearly pure carbon. They owe their forms to the conditions under which they were created or to the geological changes which they have undergone. With each succeeding stage, the percentage of carbon increases and the oxygen content decreases. The chemical changes are approximately as given in Table 1.

Origin of Coal: Combustion, Nov., 1922, p. 284.

10. Composition of Solid Fuels. — All solid fuels, when separated into their ultimate chemical constituents, are composed principally of varying proportions of carbon, hydrogen, oxygen, sulphur, and refractory earths. Carbon and hydrogen are the only desirable elements from a combustion standpoint, and the others may be considered impurities. The various combinations into which the carbon, hydrogen and oxygen are united are extremely complex and greatly influence the physical characteristics of the fuel. Not all of the carbon and hydrogen is available for combustion, since part of the carbon may be present as a carbonate and part of the hydrogen as water. The real test of any fuel is its performance under service conditions; but a knowledge of the physical and chemical characteristics, as determined in the laboratory, is of great importance in selecting the equipment best suited for the combustion of that particular fuel. The analyses most commonly used in this connection are known as " proximate " and " ultimate."

Proximate Analysis. This analysis enables the engineer to predict, to a certain extent, the behavior of the fuel in the furnace, by giving the percentages of moisture, ash, fixed carbon and volatile matter. The sulphur content and the calorific value of the fuel are usually included in the commercial proximate analysis.

TABLE 1

PROGRESSIVE CHANGE FROM PURE CELLULOSE TO GRAPHITIC ANTHRACITE

(Moisture, Ash and Sulphur Free)

Substance	Carbon	Oxygen	Hydrogen	Nitrogen
Pure cellulose	44.5	49 4	6.1	..
Wood	50 0	43.0	5.7	1.3
Peat	60 0	32 5	5 5	2.0
Lignite	66 5	26 5	5 5	1.5
Sub-bituminous coal	75.0	18 0	5 5	1.5
Bituminous coal	82.0	11.0	5 4	1.6
Semi-bituminous coal	86.5	7.0	5.0	1.5
Semi-anthracite coal	91 0	4.0	3.5	1.5
Anthracite	93.2	3 0	2 5	1.3
Graphitic anthracite	97 3	2 1	0.5	0.1

There is no definite demarcation between the analyses of the various fuels as indicated in this table, since considerable overlapping exists, but the progressive change is substantially as shown.

Moisture, as obtained from this analysis, is purely an arbitrary quantity, based upon the loss in weight of a sample when maintained for approximately one hour at a temperature of 220 deg. fahr. The material driven off in this manner is not all water, since some of the volatile combustible may distill off; furthermore, all of the water may not be evaporated by this treatment. Nevertheless, the treatment accomplishes its purpose, which is to bring the material to a condition which can be duplicated closely and represents a fixed basis for comparison. Moisture not only increases the cost of transporting and handling the fuel but is also a disadvantage in the furnace, absorbing heat which might otherwise be available for generating steam. Solid fuel free from "moisture" is known as **dry fuel.**[1]

The material which remains after the fuel has been completely burned is classified as **ash.** The term "ash," as commonly used in steam boiler practice, differs from ash as determined in the laboratory in that it contains some combustible. A better term for the former is **refuse.** It is derived from the inorganic matter in the fuel, such as sand, clay, shale, "slate," and iron pyrites, and is composed largely of compounds of silica, alumina, iron, and lime, together with small quantities of magnesia. A large percentage of ash is undesirable, since it reduces the heat value of the fuel, increases the cost of transportation and handling, necessitates disposal of refuse, and often produces troublesome clinker. Solid fuel, free from moisture and ash, is commonly designated as **combustible,** though the nitrogen and oxygen included are not combustible. Low-grade fuels are considered as such chiefly because of their large moisture and ash content.

That portion of the carbon, combined with hydrogen, and other gaseous compounds, which is driven off the dry fuel by the application of heat, constitutes the **volatile combustible matter,** or simply **volatile** matter. The term "volatile com bustible" is a misnomer, since a considerable

[1] "Moisture," as determined from the proximate analysis, must not be confused with "air-drying loss." The primary purpose of air-drying is to reduce the moisture content to such a condition that there will not be rapid changes in the weight of the sample during the course of analysis; it simply shows the amount of moisture removed in order to bring the sample to a condition of equilibrium with respect to the moisture in the air of the room. "Air-drying loss" is the amount of moisture driven off when the sample, as received, is subjected to a temperature of 86 to 95 deg. fahr. The drying process is continued until the loss in weight between two successive weighings, made six to twelve hours apart, does not exceed 0.2 per cent. See "Analysis of Coal in the United States," Bulletin 22, 1913, Bureau of Mines.

fraction of the distilled gases consists of water vapor, carbon dioxide, nitrogen, and other inert, non-combustible diluents. The importance of the " volatile matter " to the engineer is obvious, since a high percentage indicates that special care must be observed in effecting smokeless combustion.

The uncombined carbon, or that portion which remains after the volatile matter has been driven off, is known as **fixed carbon.** Fixed carbon, however, is not pure carbon, since the carbonized residue contains, in addition to the ash-forming constituents, small amounts of hydrogen, oxygen, and nitrogen, and approximately half the original sulphur content. " Fixed carbon " is a measure of the relative coking properties of coals, though in the commercial manufacture of coke or gas the yield of coke is several per cent higher than that obtained in the laboratory. In the proximate analysis of fuel, the sulphur is included in the volatile matter, fixed carbon and ash. Sulphur occurs in coal as pyrites, sulphates of iron, lime, and aluminum, and in combination with the coal substance as organic compounds. Although classed as an impurity, sulphur has a heating value, when in the form of iron pyrites, of almost one-half that of the carbon it replaces. For steaming purposes, sulphur is objectionable only when its presence produces a badly clinkering ash, or brings about corrosion by the formation of acid with moisture, as in connection with economizer installations.

ULTIMATE ANALYSIS. In the ultimate analysis, the composition of the fuel is expressed in terms of its elementary constituents of carbon, hydrogen, oxygen, nitrogen and sulphur, and ash. The ultimate analysis, while little used in ordinary practice, is of value in determining the more important heat losses incident to combustion, but an accurate analysis requires considerable time for its consummation and necessitates the services of a competent chemist. For that matter, an accurate proximate analysis requires even more skill than the ultimate analysis, since in the latter the determination of hydrogen, carbon, and nitrogen is not subject to the arbitrary conditions that must be maintained in the proximate analysis. But as ordinarily made the latter requires little apparatus and is within the skill of the average engineer.

Both the ultimate and proximate analyses may be expressed in terms of

(1) " Fuel as received," or **fuel as fired**
(2) " Fuel, moisture free," or **dry fuel**
(3) " Fuel, moisture and ash free," or **combustible**
(4) " Fuel, moisture, ash, and sulphur free."

A.S.M.E. Test Code for Solid Fuels: Mech. Engrg., Sep. 1924, p. 558.

Standard Methods of Laboratory Sampling and Analysis of Coal: Proc. Am. Soc. Testing Materials, 1921, p. 760.

Sampling Coal: U. S. Bureau of Mines, Tech. Paper 1, 1911; 133, 1917; 76, 1911.

In the various fuel publications issued by the Bureau of Mines and the U. S. Geological Survey, the quoted terms are used almost exclusively, whereas in the Boiler Code advocated by the American Society of Mechanical Engineers and in most engineering literature, the terms in bold type are given preference. Engineers prefer to have the results based on fuel as fired, since this represents the condition of the fuel as fed to the furnace. For convenience in comparing analyses, the results are usually based on dry and combustible; but occasionally, as will be shown later, the "fuel, moisture, ash, and sulphur free" basis is of service. Analyses are readily converted from one basis to another, as will be seen from the following example.

Example 1. — Given the proximate and ultimate analyses of a sample of bituminous coal "as received." Transfer these analyses to the "moisture free" and "moisture and ash free" basis. Also transfer the ultimate analysis as received to the "moisture, ash and sulphur free" basis.

Solution. —

FOR THE PROXIMATE ANALYSIS:

	Coal as Received, or Coal as Fired A.	Coal, Moisture Free, or Dry Coal B.	Coal, Moisture and Ash Free, or Combustible C
Fixed carbon	50 19	54 42	61 49
Volatile matter	31 44	34 08	38 51
Ash	10 61	11 50	. .
Moisture	7 76		.
	100 00	100 00	100 00

Column A = laboratory analysis.
Column B = column A ÷ (1 − proportional weight of moisture)
= column A ÷ 0.9224.
Column C = column A ÷ [1 − (proportional weight of moisture + ash)]
= column A ÷ 0.8163.

FOR THE ULTIMATE ANALYSIS:

	Coal as Received		Coal, Moisture Free	Coal, Moisture and Ash Free	Coal, Moisture, Ash and Sulphur Free
	A.	A_1	B.	C.	D
Carbon	66 55	66.55	72 15	81 52	83 54
Hydrogen	5.14	4 28	4 64	5 24	5.37
Nitrogen	1 32	1 32	1 43	1.62	1.66
Oxygen	14 41	7 51	8 14	9 21	9.43
Sulphur	1 97	1 97	2 14	2 41	. . .
Ash	10 61	10 61	11.50	. . . .	. . . . .
Free moisture	.	*7 76	. . .	. .	. . . . .
	100 00	100 00	100.00	100.00	100 00

* From the proximate analysis

In the ultimate analysis of the coal as received (Column A), the free moisture or "moisture" is included in the hydrogen and oxygen. Since the water is composed of one part hydrogen and eight parts oxygen, one-ninth of the moisture should be subtracted from the hydrogen and eight-ninths from the oxygen, in order to include free moisture as a separate item, thus:

Column A = laboratory analysis.

$$\begin{aligned}
\text{Hydrogen (column } A_1) &= \text{hydrogen (column A)} - \tfrac{1}{9} \times \text{per cent moisture} \\
&= 5.14 - \tfrac{1}{9} \times 7.76 \\
&= 4.28. \\
\textbf{Oxygen} \text{ (column } A_1) &= \text{oxygen (column A)} - \tfrac{8}{9} \times \text{per cent moisture} \\
&= 14.41 - \tfrac{8}{9} \times 7.76 \\
&= 7.51. \\
\text{Column B} &= \text{column } A_1 \div (1 - \text{proportional weight of moisture}) \\
&= \text{column } A_1 \div 0.9224. \\
\text{Column C} &= \text{column } A_1 \div [1 - \text{proportional weight of (moisture + ash)}] \\
&= \text{column } A_1 \div 0.8163. \\
\text{Column D} &= \text{column } A_1 \quad [1 - \text{proportional weight of (moisture + ash + sulphur)}] \\
&= \text{column } A_1 \quad 0.7966.
\end{aligned}$$

The term **free hydrogen,** or **available hydrogen,** is based on the assumption that all of the oxygen in the coal is combined with hydrogen in the proper ratio to form water, or

Free hydrogen = Total hydrogen − oxygen/8 = H − O/8.

All of the oxygen + O/8 is the weight of the **combined moisture,** and the sum of the **free moisture** (moisture as determined from the proximate analysis) and combined moisture is designated as the **total moisture.**

Example 2. — Determine the free hydrogen, combined moisture, and total moisture for coal as fired, the analysis of which is given in Example 1.

Solution. —

$$\begin{aligned}
\text{Free hydrogen} &= H - O/8 = 4.28 - 7.51/8 \\
&= 3.34. \\
\text{Combined moisture} &= O + O/8 = 7.51 + 7.51/8 \\
&= 8.45. \\
\text{Total moisture} &= M + (O + O/8) = 7.76 + 8.45 \\
&= 16.21.
\end{aligned}$$

For most engineering purposes, extreme accuracy is not necessary in determining the ultimate analysis, since the average commercial heat balance is in itself only approximate at the best. Consequently, recourse may be had to empirical formulas for approximating the weight of the chemical constituents from the proximate analysis, thus:[1]

[1] "Experimental Engineering," Carpenter and Diederichs. John Wiley & Sons, Inc., 1911, p. 507.

$$\text{For hydrogen, } H = V\left(\frac{7.35}{V + 10} - 0.013\right) \tag{1}$$

in which

H = the per cent of hydrogen in the combustible,
V = the per cent of volatile matter in the combustible.

For nitrogen,

$$N = 0.07\,V \text{ for anthracite and semi-anthracite}$$
$$= 2.10 - 0.012\,V \text{ for bituminous and lignite.} \tag{2}$$

For total carbon (fixed carbon + volatile carbon),

$$\begin{aligned} C &= F + 0.02\,V^2 \text{ for anthracite} \\ &= F + 0.9\,(V - 10) \text{ for semi-anthracite} \\ &= F + 0.9\,(V - 14) \text{ for bituminous coals} \\ &= F + 0.9\,(V - 18) \text{ for lignites} \end{aligned} \tag{3}$$

in which

C = per cent of total carbon in the combustible,
F = per cent of fixed carbon as determined from the proximate analysis,
V = as above.

Sulphur in the coal increases the value of V; hence the calculated value of C is too high by practically the sulphur content of the combustible.

Example 3. — Calculate the ultimate analysis from the proximate analysis of the coal given in Example 1.

Solution. — Substitute the various numerical values in equations (1) to (3) and solve, thus:

$$H = 38.51\left(\frac{7.35}{38.51 + 10} - 0.013\right) = 5.33 \text{ per cent. (Analysis gives } H = 5.24).$$

$$\begin{aligned} N &= 2.10 - 0.012 \times 38.51 \\ &= 1.64 \text{ per cent. (Analysis gives } N = 1.62). \\ C &= 61.49 + 0.9\,(38.51 - 14) \\ &= 83.55 \text{ per cent. (Analysis gives } C = 81.52). \end{aligned}$$

The ultimate analysis of the **coal as received,** neglecting the sulphur, is:

	Calculated Values, Per Cent	Actual Values, Per Cent
H = 5.33 } × 0.8163	4 35	4 28
N = 1.64 } × 0.8163	1 33	1 32
C = 83.55 } × 0.8163	68 20	68 52*
Ash (by analysis)	10 61	10.61
Moisture (by analysis)	7 76	7 76
O (by difference)	7 75	7.51
	100.00	100.00

* Carbon + Sulphur = 66.55 + 1.97 = 68.52.

It will be seen that the agreement is fairly close, with the exception of the figure for total carbon. As previously stated, this is largely due to the fact that the sulphur content is practically all added to the total carbon. If the sulphur content of the coal is known, as in this case (2.41 per cent), correction can be made so that the final computed value for the total carbon is 83.55 − 2.41 = 81.14 per cent per lb. of combustible.

This method of calculating the ultimate from the proximate analysis gives fairly accurate results for most coals, but with some grades of bituminous coals the results for H and C may be in error as much as 5 per cent for each constituent.

As the average plant is not equipped with the necessary apparatus for making the proximate analysis, to say nothing of the ultimate analysis, the preceding calculations are of little value to the engineer in charge. The proximate analysis is too cumbersome, ever for the large plant, when a number of heat balances are required in a short time, as when new fuels are being tried out. In such cases, the following method enables the engineer to approximate the ultimate analysis with sufficient accuracy for most practical purposes, provided the source of coal supply is known:[1]

Bulletins Nos. 22, 85 and 123, issued by the Bureau of Mines, contain a large number of ultimate analyses of coals from all parts of the country. A study of the data will show that *coals from any given bed have practically the same analysis when expressed on a "free from moisture, ash and sulphur" basis;* hence it is principally a question of determining the amount of free moisture and ash in the sample (a comparatively simple test) and in assuming the sulphur content. Since the percentage of sulphur is not uniform, some error may be introduced in making this assumption, but it is negligible as far as the average commercial heat balance is concerned. This method of obtaining the ultimate analysis is best illustrated by an example.

Example 4. — Assume that a sample of Illinois coal (analysis as per Example 1) is available, and that only the ash and moisture determinations have been made. Approximate the ultimate analysis from the average "moisture, ash, and sulphur free" analysis of Illinois coals.

Solution. — The average of a number of Illinois coals,[2] as recorded in the Government bulletins referred to, is:

Combined Moisture	Free Hydrogen	Carbon	Nitrogen
11.94	4.14	82 4	1.52

[1] P. W. Evans, *Armour Engineer*, May, 1915, p. 301.

[2] Moisture, ash, and sulphur free.

Assuming the per cent of sulphur in the coal under consideration to be the average of Illinois coals as recorded in the Government bulletins ($S = 2.84$ per cent), the total free moisture, ash, and sulphur would be $7.76 + 10.61 + 2.84 = 21.2$ per cent; and the "free from moisture, ash, and sulphur" content, $100 - 21.2 = 78.8$ per cent. The ultimate analysis of the coal as received may then be calculated as follows:

	Calculated Values, Per Cent	Actual Values Per Cent
Combined moisture, 11.94×0.788	9 40	8 45
Free moisture (by test)	7 76	7.76
Free hydrogen, 4.14×0.788	3 26	3 34
Total carbon, 82.4×0.788	64 98	66 55
Nitrogen, 1.52×0.788	1 19	1 32
Ash (by test)	10 61	10 61
Sulphur	*2 80	1.97
	100 00	100 00

* By assumption.

The agreement between calculated and actual values for most Illinois coals is much closer than in this particular example. The splendid work of the U. S. Bureau of Mines has placed at the disposal of the public complete analyses of the coals of all the coal fields in the country, and the error in assuming the average values of an entire state, as in the preceding example, may be greatly reduced by taking the average values for the particular field in which the coal under consideration is mined.

11. Coal. — Coal is the most important of all fuels and furnishes the greater part of the world's heat and power energy. According to the latest estimates, the coal reserves of the world, by continents, are as follows:

	Billions of Tons (2000 lb.)
America	5,628
Asia	1,410
Europe	864
Oceania	188
Africa	61

Of the amount contained in the Americas, the United States claims 4,205 billion tons, or 51 per cent of the total coal of the world. The present (1924) rate of production in the United States is approximately 700 million tons per annum and the distribution is roughly as follows:

	Per Cent		Per Cent
Industrial steam trade	33	Exports	4
Railroad fuel	28	Steamship bunkers at tidewater	2
Domestic and small trade	16	Used at mine for steam and heat	2
Manufacture of coke	9	Manufacturer of coal gas	1
Manufacture of by-product cokes	4		

The greatest repository of coal in the United States is west of the Mississippi River, but almost 80 per cent of the present production is from the States east of this river. Some idea of the extent and character of these fields may be gained from an inspection of the chart in Fig. 8.

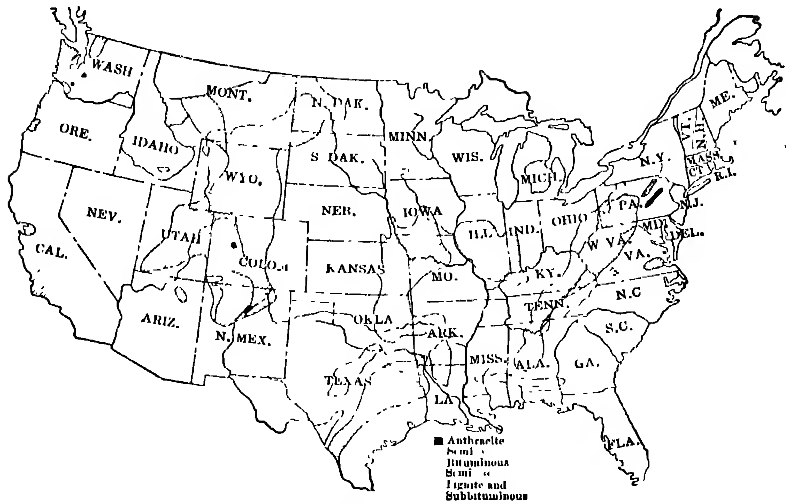

FIG. 8. Coal Map of the United States.

Coals and allied substances have been variously classified, according to

1. Oxygen-hydrogen ratio, or Gruner's classification.
2. Fixed carbon and volatile combustible matter.
3. Fuel ratio, or the ratio of the fixed carbon to the volatile combustible matter.
4. Calorific power.
5. Fixed carbon.
6. Total carbon.
7. Hydrogen.
8. Carbon-hydrogen ratio, or the ratio of the total carbon to the hydrogen.

All of these classifications are more or less unsatisfactory because of the overlapping of the various groups.

According to the investigations of Prof. S. W. Parr, the ratio of heat value to the percentage of volatile matter of pure coal (moisture, ash, and sulphur free) is a more reliable means of classifying coals and allied substances than any of those previously mentioned.

In its various bulletins the U. S. Bureau of Mines uses the following classification for the **ranks** of coal.

Anthracite
Semi-anthracite
Semi-bituminous
Bituminous
Sub-bituminous
Lignite

The word "rank" in this connection is used to designate "those differences in coal that are due to the progressive change from lignite to anthracite, a change marked by the loss of moisture, of oxygen, and of volatile matter." This change is generally accompanied by an increase of fixed carbon, of sulphur, and probably of ash. When, however, one coal is distinguished from another by the amount of ash or sulphur it contains, this difference is said to be one of **grade.** Thus, a "high-grade coal" means merely one that is relatively pure, whereas a "high-rank coal" means one that is high in the scale of coals, or in other words, one that has suffered devolatilization and that now contains a smaller percentage of volatile matter, oxygen, and moisture than it contained before the change occurred. M. R. Campbell (Prof. Paper 100-A. U. S. Geological Survey, 1917) gives the following analyses as representative of the different ranks of coal, computed on nine samples as received, on the ash-free basis.

	Fixed Carbon	Volatile Matter	Moisture	Heat Value B t u. per lb.
	Per Cent by Weight			
Lignite	37 8	18 8	43 4	7,400
Sub-bituminous	42 1	34 2	23 4	9,720
Bituminous				
Low rank	47 0	41 4	11 6	12,880
Medium rank	54 2	40 8	5 0	13,880
High rank	64 6	32 2	3 2	15,160
Semi-bituminous				
Low rank	75 0	22 0	3 0	15,480
High rank	83 4	11 6	5.0	15,360
Semi-anthracite	83 8	10.2	6 0	14,880
Anthracite	95 5	1.2	3 2	14,440

The U. S. Geological survey used the following names for designating the areas underlain by coal-bearing rocks.

Coal district is the term applied only to small coal areas in which mines are developed continuously on a given bed or beds and the coal is generally known by some distinguishing feature, such as a trade name, or by some physical characteristic upon which it is advertised or sold. Districts are generally named from the leading town in the county or from the town at which mining first achieved distinction in producing this particular kind of coal. Examples of districts are the Red Lodge district of Mon-

tana, the Gallup district of New Mexico, and the Winfield district of West Virginia.

The term **coal field** is applied to an area generally larger than a district but still well-defined and compact. Small areas or basins that are separated from the main coal are called fields, especially if this coal is of fairly uniform composition and value. Examples of coal fields are the Pocahontas field of Virginia and West Virginia, the New River field of West Virginia, and the Windber field of Pennsylvania.

Coal fields are grouped into larger divisions called **regions**. Such grouping is generally designed to bring together coal fields that have some feature or features in common, thus enabling them to be considered as a whole or separately as the problem may demand. Good examples are the anthracite region of Pennsylvania, the western coal region in Iowa, and numerous deep basins of the Rocky Mountain States.

As fields are grouped into regions, so regions are grouped into much larger divisions, called **provinces.** These are the Eastern province, Interior province, Gulf province, Northern Great Plains province, Rocky Mountain province, and Pacific Coast province.

For a detailed description of the various districts, fields, regions, and provinces in the United States, consult "The Coal Fields of the United States," by M. R. Campbell, U. S. Geological Survey, Prof. Paper 100-A, 1917.

Analysis of Coals in the United States: U. S. Bureau of Mines, Bulletins, No. 193, 1922; No. 123, 1918; No. 85, 1914; No. 22, 1913; Technical Paper 76, 1914.

12. Anthracite. — Anthracite, commonly known as hard coal, consists almost entirely of fixed carbon and is the hardest of all the coals. Specific gravity 1.4 to 1.6; fuel ratio not more than 50 or 60 and not less than 10. It has a deep black color, a shiny, semi-metallic luster, has few points and clefts, and burns without softening or swelling. It ignites slowly and burns at a high temperature with little flame or smoke. As nearly all anthracite, with some unimportant exceptions, comes from three small fields in Eastern Pennsylvania, the supply is comparatively limited (estimated at less than 5 per cent of the total unmined reserve coal in the United States). Anthracite, as marketed, is always "sized" or screened, but there is no accepted standard of sizes, each coal district having certain sizes and names peculiar to itself and to the trade it supplies. Table 4 gives one of the standard divisions of mesh and the trade names under which it is classed and marketed. The price of the finer sizes is much less than that of the coarser, partly because of the premium placed on the larger sizes by the demand for domestic heating. Even in the immediate vicinity of the mines, sizes over "pea coal" are usually

prohibitive in price for steam power plant use. The smaller sizes are quite commonly used in city plants where smoke ordinances are rigidly enforced, and when the price compares favorably with other available grades. **Culm,** or the refuse from screening, and **bone coal** (that part of the material encountered in mining which contains a large percentage of coal but not of marketable grade) were formerly rejected to waste on account of their high ash content and low heating value; but with the increasing cost of coal and the improvement of furnace design, nearly all of this refuse is being made available for power plant use. The proximate and ultimate analyses of a number of anthracites are given in Table 2. It is believed by many that anthracite has greater heat value than any of the other ranks, but this is not true, as will be seen by a comparison of the analyses in Table 2.

Use of River Coal at Baltimore. Combustion, Nov., 1923, p. 388.

13. Semi-anthracite. — Semi-anthracite kindles more readily and burns more rapidly than anthracite. It requires little attention, burns freely with a short flame, and yields great heat with little clinker and ash. It is apt to split on burning and wastes somewhat in falling through the grate. It swells considerably but does not cake. Semi-anthracite has less density, hardness, and metallic luster than anthracite and can generally be distinguished from pure anthracite by its tendency to soil the hands. Semi-anthracite is not of great importance in the steam power plant field on account of the limited supply and high cost. It is mined chiefly in a few small areas in Pennsylvania, Arkansas, and Virginia. Some excellent deposits have also been found in Alaska. Specific gravity 1.3 to 1.4; fuel ratio 6 to 10. See Table 2 for analyses of a few typical specimens.

14. Semi-bituminous. — Semi-bituminous is similar in appearance to semi-anthracite, but it is softer and contains more volatile matter (15 to 22 per cent). It has a high heating value, low moisture, ash, and sulphur content, burns freely without producing objectionable smoke and ranks among the best steaming coals in the world. The volatile matter in semi-bituminous coals is of remarkably uniform composition and approaches methane (CH_4) in its analysis. While semi-bituminous coal is found in several states, the chief supply comes from the Pocahontas and New River fields of Virginia and West Virginia, the Georges field of Maryland, the Windber field of Pennsylvania, and the western end of the Arkansas field. The supply of semi-bituminous coal is comparatively limited, and it probably will be the first to be exhausted because it has a greater efficiency and is adapted to more diverse uses than anthracite. Practically all semi-bituminous coals are of the caking variety, and some of

TABLE 2

ANALYSES OF REPRESENTATIVE COALS OF THE UNITED STATES

(Compiled from *Bureau of Mines Bulletins*)

Run of Mine — As Received

State	County, Field, District, or Trade Name		Proximate Analysis: Moisture	Volatile Matter	Fixed Carbon	Ash	Ultimate Analysis: Sulphur	Hydrogen	Carbon	Nitrogen	Oxygen	Heating Value B.t.u. per lb.: As Received	Moisture and Ash Free
	Anthracite												
Alaska	Bering River	F	3.65	9.20	84.58	2.57	0.60	3.20	86.49	1.11	6.03	14,182	15,124
Col.	Crested Butte	F	2.70	3.32	88.15	5.83	0.80	3.28	85.38	1.12	3.59	14,099	15,413
N. M.	Cerillos	F	5.70	2.18	86.13	5.99	0.69	2.38	82.87	1.26	6.81	13,268	15,025
Pa.	Lackawanna	C	3.43	6.79	78.25	11.53	0.46	2.52	78.85	0.77	5.87	12,782	15,030
Pa.	Luzerne	C	1.31	5.68	85.87	7.14	0.42	2.35	86.76	0.68	2.65	13,777	15,048
Wash.	Whatcom	C	4.36	7.45	75.96	12.23	0.96	2.97	77.75	0.98	5.11	12,593	15,098
	Semi-anthracite												
Alaska	Bering River	F	1.18	10.02	84.50	4.30	1.59	3.23	86.37	1.46	3.05	14,344	15,176
Ark.	Pope	C	2.79	11.90	75.24	10.07	2.17	3.69	78.28	1.66	4.13	13,356	15,327
Pa.	Bernice	D	3.16	8.59	78.08	10.17	0.67	3.47	79.49	1.10	5.10	13,376	15,431
Va.	Montgomery	C	1.67	9.36	66.65	22.32	0.71	3.19	69.24	0.81	3.73	11,570	15,223
	Semi-bituminous												
Ala.	Lookout Mt.	F	3.38	18.67	63.41	14.54	1.22	4.29	72.86	1.27	5.82	12,701	15,475
Alaska	Bering River	F	3.63	15.37	73.95	7.05	1.18	3.84	77.85	1.64	8.44	13,116	15,363
Ark.	Logan	C	2.77	14.69	73.47	9.07	2.79	4.02	78.71	1.46	3.95	13,774	15,624
Col.	Coal Basin	F	3.07	22.67	65.10	9.16	0.63	4.96	78.81	1.69	4.75	13,990	15,939
Ga.	Mann	D	3.80	15.88	65.83	14.49	1.27	4.32	70.59	1.09	8.24	12,791	15,653
Md.	Georges Cr.	F	3.40	15.00	75.10	6.50	1.04	4.63	80.69	1.55	5.60	14,160	15,710
Okla.	Le Flore	C	2.63	16.48	72.22	8.67	1.00	4.25	80.06	1.72	4.30	13,799	15,556
Pa.	Windber	F	2.54	19.85	71.22	6.39	2.12	4.70	81.17	1.33	4.29	14,315	15,719
Pa.	Broad Top	F	2.14	15.47	75.96	6.43	1.05	4.44	82.83	1.27	3.98	14,470	15,827
Va.	Pocahontas	F	1.63	17.17	75.34	5.86	0.75	4.58	83.14	1.02	4.65	14,672	15,860
W. Va.	New River	F	3.34	21.25	73.18	2.23	0.56	5.13	84.19	1.55	6.34	14,821	15,696
W. Va.	Pocahontas	F	3.61	17.41	74.84	4.14	0.76	4.57	83.68	1.12	5.73	14,587	15,815
	Bituminous — Cannel												
Ky.	Eastern	F	1.70	50.76	38.23	9.31	1.02	6.83	73.25	1.31	8.28	14,251	16,013
W. Va.	Kanawha	F	1.80	44.90	49.86	3.44	0.87	6.96	80.57	1.51	6.65	15,330	16,176
Utah	Kane	C	7.35	46.93	22.48	23.24	1.61	6.18	51.88	1.06	16.03	10,355	14,918
	Bituminous												
Ala.	Cahaba	F	2.85	36.80	53.77	6.58	0.49	5.14	75.82	1.10	10.89	13,390	14,783
Ala.	Warrior	F	1.66	30.43	63.29	4.62	1.40	5.13	81.53	1.55	5.77	14,605	15,584
Alaska	Chignik Bay	D	7.06	31.48	39.68	21.78	1.30	4.83	55.14	0.61	16.34	9,846	13,838
Alaska	Matanuska	F	2.18	30.60	58.06	9.16	0.70	4.83	71.43	1.50	12.38	13,145	14,827
Calif.	Stone Canyon	D	6.95	46.69	40.13	6.23	4.17	6.28	66.01	1.17	16.14	12,447	14,336
Col.	Canyon City	F	11.19	36.77	45.75	6.29	0.92	5.44	62.50	0.96	23.89	11,280	13,676
Col.	Trinidad	F	2.33	25.82	54.58	17.27	0.52	4.62	69.14	1.07	7.38	12,337	15,345
Col.	Durango	F	4.84	34.56	42.30	18.30	7.56	4.85	59.90	1.11	8.28	11,065	14,396

F Field C County D District

TABLE 2 — *Continued*

State	County, Field, District, or Trade Name	Proximate Analysis: Moisture	Proximate Analysis: Volatile Matter	Proximate Analysis: Fixed Carbon	Proximate Analysis: Ash	Ultimate Analysis: Sulphur	Ultimate Analysis: Hydrogen	Ultimate Analysis: Carbon	Ultimate Analysis: Nitrogen	Ultimate Analysis: Oxygen	Heating Value B t u. per lb.: As Received	Heating Value B t u. per lb.: Moisture and Ash Free
	Bituminous — Continued											
Col.	Yampa F	8.85	36.06	50.19	4.90	1.51	5.69	68.59	1.54	17.77	12,161	14,099
Idaho	Fremont C	11.45	37.24	47.01	4.30	0.54	5.94	68.09	1.40	19.73	12,094	14,357
Ill.	Clinton C	11.35	34.62	40.63	13.40	4.76	5.41	57.36	1.05	18.02	10,733	14,263
Ill.	Franklin C	9.93	33.13	48.77	8.17	0.75	5.46	67.08	1.41	17.13	11,896	14,522
Ill.	La Salle C	12.39	36.89	41.80	8.92	3.92	5.85	61.29	1.00	19.02	11,399	14,486
Ill	Marion C	11.93	29.99	43.90	14.18	4.29	5.21	56.94	1.01	18.37	10,303	13,943
Ill.	Saline C	7.81	33.54	50.27	8.38	2.36	5.31	67.40	1.44	15.11	12,418	14,818
Ill	Williamson C	8.20	32.26	46.59	12.95	3.48	5.09	62.52	1.10	14.86	11,362	14,409
Ind.	Clay (Block) C	16.91	26.85	38.87	17.37	1.89	5.48	52.97	1.01	21.28	9,524	14,492
Ind.	Greene (Block) C	13.58	32.07	46.20	8.15	0.91	5.65	63.53	1.42	20.34	11,419	14,589
Ind.	Knox C	10.72	37.17	42.77	9.34	1.24	5.45	63.63	1.25	16.10	11,615	14,530
Ind.	Sullivan C	13.05	34.20	47.61	5.04	0.82	6.07	67.34	1.41	19.29	12,022	14,679
Iowa	Appanoose C	14.08	35.59	39.37	10.96	4.26	5.57	58.49	0.90	19.82	10,723	14,305
Iowa	Lucas C	15.39	30.19	41.49	12.63	3.19	5.74	55.81	1.14	21.50	10,242	14,229
Iowa	Polk C	13.88	36.94	35.17	14.01	6.15	5.52	54.68	0.84	18.80	10,244	14,206
Kans.	Cherokee C	2.50	33.80	51.25	12.45	5.68	4.91	69.07	1.20	6.69	12,900	15,167
Kans.	Crawford C	4.85	33.53	52.52	9.10	4.95	5.08	71.20	1.21	8.43	12,942	15,041
Kans.	Linn C	9.04	29.69	45.55	15.72	3.72	5.01	60.99	1.06	13.50	11,142	14,809
Ky.	Eastern F	3.56	37.18	56.55	2.71	0.70	5.54	79.61	1.61	9.83	14,220	15,172
Ky.	Western F	9.48	35.65	44.92	9.95	3.41	5.47	64.72	1.42	15.03	11,533	14,314
Ky.	Western F	4.37	36.27	47.67	11.69	3.58	5.04	68.26	1.44	9.99	12,325	14,083
Md.	George's Crk F	3.10	15.50	74.50	6.90	0.86	4.57	80.71	1.69	5.32	14,070	15,630
Md.	Potomac F	2.38	15.96	70.20	11.51	2.53	4.31	76.77	1.25	3.63	13,343	15,489
Mich.	Saginaw D	11.91	31.50	49.75	6.84	1.24	5.84	66.56	1.19	18.33	11,781	14,409
Mo.	Barton C	5.87	30.98	51.67	11.48	5.00	5.07	67.64	1.09	9.72	12,339	14,029
Mo.	Macon C	13.81	34.69	41.79	9.71	3.32	5.71	59.84	1.06	20.36	10,964	14,341
Mo	Vernon C	6.50	32.61	50.83	10.06	4.95	5.21	67.97	1.09	10.72	12,458	14,931
Mont.	Bridger D	8.56	32.26	45.69	13.49	0.54	5.06	60.39	1.06	19.56	10,685	13,689
Mont.	Lewiston D	12.59	26.71	43.46	17.24	3.51	4.84	54.83	0.76	18.82	9,396	13,390
Mont	Livingston D	6.26	30.60	36.88	26.26	0.68	4.77	53.79	0.86	13.64	9,792	14,510
N. M	Raton F	3.64	36.10	48.26	12.00	0.70	5.25	69.76	1.31	10.98	12,623	14,963
N. M.	Socorro C	6.48	34.51	51.92	7.09	0.50	5.32	69.35	1.17	16.57	11,992	13,874
Ohio	Belmont C	2.97	37.61	49.45	9.97	3.65	5.14	70.21	1.23	9.80	12,935	14,855
Ohio.	Guernsey C	6.49	35.41	52.57	5.53	0.88	5.49	73.41	1.37	13.32	12,940	14,708
Ohio	Hocking C	9.72	32.44	53.41	4.43	0.54	5.70	69.50	1.25	18.58	12,247	14,269
Ohio	Jefferson C	3.50	37.98	51.08	7.44	3.09	5.43	73.39	1.46	9.19	13,286	14,918
Okla.	Coal C	6.24	37.26	43.29	13.21	3.96	4.93	62.34	1.36	14.20	11,228	14,039
Okla.	Haskell C	2.70	21.07	69.88	6.35	0.77	4.46	81.33	1.67	5.42	14,098	15,502
Okla.	McAlester F	3.53	38.20	51.74	6.53	1.22	5.40	75.93	2.03	9.09	13,559	15,077
Pa	Allegheny C	2.73	36.03	54.98	6.26	1.39	5.26	76.82	1.46	8.81	13,815	15,181
Pa.	Cambria C	1.54	23.86	67.08	7.52	1.53	4.73	80.47	1.38	4.37	14,195	15,608
Pa	Fayette C	5.13	27.87	58.29	8.71	0.86	4.91	73.13	1.50	10.89	13,365	15,511
Pa	Indiana C	1.30	26.70	64.40	7.60	2.03	5.02	80.35	1.36	3.64	14,315	15,714
Pa	Jefferson C	2.44	28.44	60.68	8.44	1.32	5.07	76.91	1.31	6.95	13,732	15,410
Pa	Washington C	1.96	30.55	58.24	9.25	2.19	4.81	74.37	1.45	7.93	13,622	15,343
Pa.	Westmoreland C	2.61	21.42	64.38	11.59	1.94	4.63	75.81	1.25	4.78	13,365	15,577
Tenn.	Anderson C	1.83	36.59	56.50	5.08	1.32	5.42	78.95	1.91	7.32	14,200	15,253
Tenn.	Campbell C	2.77	34.95	54.18	8.10	0.88	5.06	74.11	1.96	9.89	13,325	14,951

F Field C County D District

TABLE 2 — *Concluded*

State	County, Field, District, or Trade Name		Proximate Analysis: Moisture	Proximate Analysis: Volatile Matter	Proximate Analysis: Fixed Carbon	Proximate Analysis: Ash	Ultimate Analysis: Sulphur	Ultimate Analysis: Hydrogen	Ultimate Analysis: Carbon	Ultimate Analysis: Nitrogen	Ultimate Analysis: Oxygen	Heating Value B.t.u. per lb.: As Received	Heating Value B.t.u. per lb.: Moisture and Ash Free
						Bituminous — *Concluded*							
Tenn.	Fentress	C	3.40	35.15	51.53	9.92	2.64	5.15	71.63	1.46	9.20	13,050	15,055
Tenn.	Grundy	C	3.92	27.23	54.76	14.09	0.94	4.81	69.97	1.29	8.90	12,508	15,255
Utah	Carbon	C	4.06	37.99	49.89	8.06	1.15	5.66	71.91	1.47	11.75	12,836	14,607
Utah	Emery	C	7.18	42.16	44.81	5.85	0.68	5.71	69.85	1.43	16.48	12,537	14,414
Utah	Iron	C	4.93	37.24	44.79	13.04	6.72	5.11	63.01	0.93	11.19	11,412	13,012
Va.	Lee	C	4.06	34.93	56.28	4.73	1.20	5.32	76.59	1.24	10.92	13,826	15,156
Va.	Russell	C	1.84	36.15	55.48	6.53	0.54	5.25	79.28	1.49	6.91	14,098	15,383
Va.	Wise	C	2.48	31.71	60.30	5.51	0.52	5.59	79.60	1.56	7.13	14,252	15,489
Wash.	King	C	4.94	33.01	40.97	21.08	0.54	5.13	59.35	1.24	12.66	10,733	14,508
Wash.	Kittitas	C	4.37	33.21	49.48	12.94	0.35	5.46	69.83	1.56	9.86	12,524	15,147
Wash.	Pierce	C	5.94	23.17	61.13	9.76	0.41	5.09	74.04	2.17	8.53	13,165	15,617
W. Va.	Kanawha	F	5.09	29.07	62.57	3.27	1.03	5.33	78.23	1.51	10.63	14,110	15,397
W. Va.	Big Sandy	F	2.75	34.91	56.68	5.66	1.24	5.34	78.63	1.43	7.70	14,071	15,363
W. Va.	Wheeling	F	3.78	37.58	50.91	7.73	1.65	5.26	73.11	1.43	10.82	13,124	14,832
W. Va.	Monongalia	D	1.63	28.42	62.01	7.94	0.96	5.00	78.24	1.28	6.58	13,937	15,313
Wyo.	Carbon	C	14.29	31.82	48.80	5.00	0.46	5.62	60.70	1.02	27.20	10,600	13,134
Wyo.	Rock Springs	F	11.64	36.37	48.58	3.41	0.81	5.72	66.08	1.43	22.55	11,768	13,855
Wyo.	Kemmerer	F	6.56	39.20	47.78	6.46	1.37	5.29	69.56	1.25	16.07	12,359	14,207
						Sub-bituminous							
Alaska	Fairhaven	D	19.74	36.25	38.84	5.17	1.21	6.26	54.82	1.03	31.51	9,583	12,762
Ariz.	Black Mesa	F	9.88	32.64	46.86	10.62	1.12	5.42	62.00	1.13	19.71	10,800	13,585
Calif	Tesla	D	18.51	35.33	30.67	15.49	3.05	5.93	47.34	0.66	27.53	8,507	12,890
Col	Boulder	C	19.15	30.82	44.27	5.76	0.25	5.93	56.38	1.08	30.60	9,616	12,807
Mont	Red Lodge	F	9.31	34.14	45.87	10.68	1.99	5.24	59.54	1.34	21.21	10,472	13,088
N. M.	Gallup	F	11.79	38.13	39.68	10.40	0.65	5.71	61.57	1.14	20.53	10,897	14,009
Ore.	Coos Bay	F	16.10	31.10	39.63	13.17	0.81	5.53	51.07	1.19	28.23	9,031	12,769
Utah	Summit	C	14.20	36.00	44.80	5.00	1.41	5.79	61.40	1.09	25.31	10,630	13,160
Wash	Pierce	C	4.11	24.38	44.75	26.76	0.44	4.31	58.16	1.39	8.94	10,204	14,891
Wyo.	Sheridan	D	29.33	28.58	36.42	5.67	1.21	6.27	45.09	0.89	40.87	7,783	11,972
Wyo.	Rock Springs	F	18.86	29.17	47.85	4.12	0.49	5.64	58.96	1.45	29.34	10,283	13,349
						Lignite							
Alaska	Igloo Creek	D	25.73	36.39	34.52	3.36	0.15	6.87	51.41	0.71	37.50	8,735	12,317
Ark.	Ouachita	C	39.43	26.49	24.37	9.71	0.49	6.98	36.33	0.68	45.81	6,356	12,497
Calif.	Ione	F	45.78	30.86	15.78	7.58	1.01	8.20	32.91	0.32	49.98	6,055	12,982
Mont	Glendive	D	34.55	35.34	22.91	7.29	1.10	6.60	42.40	0.57	42.13	7,000	12,172
N. D.	Williston	F	35.96	31.92	24.37	7.75	1.15	6.54	41.43	1.21	41.92	7,069	12,557
N. D.	Williams	C	44.17	23.79	26.28	5.76	0.56	7.28	36.11	0.63	49.66	6,035	12,055
S. D.	Perkins	C	39.16	24.68	27.81	8.35	2.22	6.60	38.02	0.53	44.28	6,307	12,017
Tex.	Houston	C	34.70	32.23	21.87	11.20	0.79	6.93	39.25	0.72	41.11	7,056	13,048
Tex.	Wood	C	33.71	29.25	29.76	7.28	0.53	6.79	42.52	0.79	42.09	7,348	12,452
						Peat (raw)							
Conn.	Beaver Marsh	D	91.20	6.61	1.84	0.35	0.02	10.59	5.00	0.13	83.91	850	10,070
Fla.	Duval	C	73.10	14.00	8.05	4.85	1.06	9.27	13.37	0.53	70.92	2,309	10,470
Mich.	Elk Marsh	D	66.91	19.04	9.29	4.76	0.09	8.97	17.23	0.81	68.14	3,024	10,700

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them furnish the best coke that is made. (See middle of paragraph 15). Specific gravity 1.3 to 1.4; fuel ratio 3 to 7. See Table 2 for analyses of a few typical specimens.

15. Bituminous. — This fuel, sometimes called **soft coal,** is the most widely distributed, and is used more extensively than any other fuel in the generation of steam. The physical properties of the different grades vary within wide limits, and no classification so far made has met with general approval. Bituminous coals range in color from pitch black to dark brown, and in hardness from that of the lignites to that of semi-bituminous. The volatile matter and fixed carbon content are about equal, but this is also true of sub-bituminous coal and lignite. One distinguishing feature which serves to separate bituminous from the lower rank coals is that of **weathering.** Bituminous coals are only slightly affected chemically by weathering, unless exposed for years; and then, although the coal consists of small particles, each particle is a prismatic fragment, whereas coals of a lower rank break into thin plates parallel with the bedding. (M. R. Campbell, Prof. Paper 100-A, U. S. Geological Survey.) Bituminous coals are either **caking** or **non-caking.** The former tend to form into a solid mass when heated in a retort or furnace, while the latter burn freely without fusing. Coals suitable for making commercial coke are called **coking** coals, but as certain grades of coke can be made from free-burning coals the term is somewhat of a misnomer. Practically all coking coals are of the caking variety, but the reverse is not necessarily true. The high-volatile coals of western Pennsylvania, eastern Ohio, eastern Kentucky, and parts of West Virginia, frequently grouped under the heading "Pittsburgh coal," possess caking qualities to a greater or less degree, but not to such an extent as the semi-bituminous coals. The non-caking variety is generally known as **free-burning** and is found chiefly in the Western and Middle Western States. Caking coal is rich in volatile hydrocarbons and is valuable in gas manufacture, and constitutes a large percentage of the steam fuel used in the Eastern States. Michigan bituminous is free-burning, but has a considerable tendency to clinker. The coals found in Illinois, Indiana, and Missouri are practically all free-burning. Iowa coals are of much lower grade than those just mentioned, because of their large moisture and ash content. Kentucky, Tennessee, and Alabama bituminous coals are high-grade and free-burning, although the coals from some localities in this district have a tendency to clinker badly. The high-volatile bituminous coals of Colorado, Wyoming, Washington, and Oregon include both caking and non-caking varieties. Specific gravity 1.2 to 1.4; fuel ratio 1 to 3.

Cannel coal is a variety of bituminous coal found in a few small areas of several states. It has the highest hydrogen content of any coal, and

burns with a bright flame without fusing. It is seldom used for steam generation, but finds a ready market because of its usefulness in the enriching of illuminating gas. Cannel coal differs greatly in appearance from all other bituminous coal, being homogeneous, with a black or grayish-black color and a dull, resinous luster.

Splint coal is a non-caking bituminous coal of singular structure and low volatile content. It splits like slate along the seams, but breaks with difficulty on cross fracture. Because of its slaty structure, low volatile content, and slow ignition, it is little used for steaming purposes.

Block coal is a variety of Indiana bituminous, laminated in structure and consisting of successive layers of coal, easily separated into thin sheets. It is used for both domestic and power plant service. See Table 2 for analyses of a number of typical specimens of bituminous coal.

Coke may be prepared from almost any fuel containing carbon, but the greater part of the commercial product comes from the distillation of bituminous coking coals. Most of the coke produced to-day is used for metallurgical and gas-making purposes, although there is a steadily increasing demand for coke for domestic heating, and, to a limited extent, for steam generation in power plants. For the latter purpose, **coke breeze** (the fine refuse from the coke ovens, quenching tables, and grading screens) is most commonly used. Dry coke is composed of practically pure carbon and ash, with small amounts of volatile matter and sulphur. Under ordinary conditions the moisture content ranges from 5 to 10 per cent, but this may be increased on exposure to as high as 25 per cent. The ash content of coke breeze varies from 10 to 35 per cent, depending upon the initial ash content of the coal used for making the coke, and the care used in preparation. Coke breeze is a low-priced, smokeless fuel, and is finding favor with engineers in large cities where smoke ordinances are rigidly enforced. It may be burned satisfactorily with forced-draft traveling-grate stokers fitted with non-sifting links, and on stationary grates of the pin-hole type using forced draft.

The Coking of Coal at Low Temperatures: Univ. of Ill., Bul. No. 30, June 3, 1912.
By-Product Coke and Coking Operation: Trans. A.S.M.E., Vol. 39, 1917, p. 897.
Metallurgical Coke: Bureau of Mines, Tech. Paper No. 50, 1913.

Smokeless fuel, manufactured from bituminous coal by a semi-coking process, has made its appearance on the market. Although the introduction of this fuel is a step toward the economic use of one of our greatest natural resources, only a small quantity of it is being produced. A typical fuel of this class, and one that demanded a great deal of attention during the war, is manufactured under the trade name of **carbocoal.**

Coal Carbonization as Applied to Power Plant Practice: Power, May 29, 1923, p. 831.
Complete Gasification of Coal. Combustion, Feb., 1923, p. 105.
Distillation Products of Coal: N.E.L.A., 1923 Report, Part A, p. 310.

16. Sub-bituminous. — This is the term adopted by the U. S. Bureau of Mines for what is commonly known as "black lignite." This class of coal is not lignitic in the sense of being woody, but closely approaches the lowest grade of bituminous in structure and in heating value. Large deposits of sub-bituminous coal are found in the Western States, principally in Colorado, Wyoming, Montana, New Mexico, Oregon, and Washington. When sub-bituminous coal is exposed to weather it slacks rapidly, the lumps becoming brittle and crumbling into fine particles. This property, in addition to the high moisture content, renders transportation unprofitable, and most of the fuel is mined for local use. Recent progress in the development of furnaces and stokers makes possible efficient combustion of this class of comparatively low-grade fuel. See Table 2 for analyses of typical samples of sub-bituminous coal.

17. Lignite, or Brown Coal, is a substance of more recent geological formation than coal, and represents a stage in development intermediate between coal and peat. Its specific gravity is low, 1.2, and when freshly mined it contains as much as 50 per cent moisture. It is non-caking, and slacks or crumbles on exposure to air. The lumps check and fall into small, irregular pieces, with a tendency to separate into extremely thin plates. Lignite deteriorates greatly during storage or long transportation. As mined, it is a low-grade fuel with a heating value of about one-half that of good coal. Vast deposits of lignite are found in Texas, Montana, the Dakotas, and Alaska. Although it ranks among the lowest grades of fuel with which the combustion engineer must work, it can be efficiently burned in the raw state in specially designed stoker-fired furnaces, or, with the usual preparation, in powdered form. When properly treated and compressed into briquettes, lignite resists weathering satisfactorily, permits handling and transportation without excessive deterioration, and is practically smokeless. See Table 2 for analyses of typical samples of lignite.

North Dakota Lignite as a Fuel for Power Plant Boilers: U. S Bureau of Mines, Bul. 2, 1910.

Briquetting Tests of Lignite. U. S. Bureau of Mines, Bul. 14, 1911.

Combustion of Lignites: Power, Apr. 8, 1919, p. 525; Dec. 16, 1919, p. 798: Combustion, Apr., 1923, p. 256.

Lignite Char, O. P. Hood, Mech. Eng'r'g., May, 1923.

18. Peat, or Turf, is nothing more than decomposed or decomposing vegetable matter containing about 90 per cent of extraneous moisture. Because of its high water content, it is unsuitable for fuel until dried. Peat is little used in this country at present, though the deposits are extensive and widely distributed, and its possibilities are beginning to attract the attention of engineers. It is estimated that there are 13 billion

tons of peat on the 20 million acres of peat-bearing lands within the United States. When properly prepared and compressed into briquettes, peat is an excellent fuel, and its adoption as a boiler fuel in this form is merely a matter of cost. Excellent results have been obtained from the combustion of peat in the pulverized form. Table 2 gives the analyses of a few typical samples of raw peat. The dry pulverized peat contains about 3 per cent moisture, 10 per cent ash, and 0.4 per cent sulphur, and has a heating value of 9000 to 10,000 B.t.u. per lb.

Peat Resources of the U. S.: Combustion, Aug., 1922, p. 70.
The Uses of Peat: U. S. Bureau of Mines, Bul. 16, 1911.
Production of Peat Fuel Combustion, Sept. 1922, p. 135.

19. Wood, Wood Waste, Tanbark, Bagasse.—Wood, as utilized commercially for steam generating purposes, is usually a waste product from some industrial process. Thus, in the vicinity of lumber camps, undesirable tree trunks, boughs and branches constitute this waste, while in sawmills and woodworking establishments the refuse material is sawdust, shavings, slabs, blocks and edgings. Chemically, there is very little difference between the various kinds of wood, but physically the variation is a wide one, particularly as regards the moisture content. The heating value of dry wood ranges from 7300 to 9900 B.t.u. per lb., and, contrary to general supposition, hard wood gives less heat than soft wood. Ordinarily, the heating value of wood is considered equivalent to 0.4 that of bituminous coal, but this is a very rough rule since the moisture content greatly influences the amount of heat available for steaming purposes. In order to produce a fuel of more uniform size and one that is more readily handled, many mills " hog " or macerate the logs, slabs and stocks. The hogged wood, mixed with the sawdust and shavings, makes a very desirable form of fuel. The moisture content varies from 20 to 60 per cent, with an average of about 45 per cent. A cord of wood equals 4 by 4 by 8 feet, or 128 cubic feet. From 55 to 75 per cent of this volume is solid wood, and the remainder interstitial spaces, the smaller value referring to sizes between 3 and 6 inches in diameter and the larger to " timber " cords. On account of loading, transportation, and storage limitations, wood waste is rarely burned, except at the mill or plant. Wood furnishes only a small part of the fuel used for power plant purposes.

Hogged Fuel Power Plant Engineering, Apr. 15, 1922, p. 407.
Burning Sawdust: Power, Dec. 31, 1921, p. 914.
Utilization of Wood Waste as Fuel in Steam Power Plants: Mech. Engrg., July, 1925, pp. 545, 550, 552.

TABLE 3

PHYSICAL AND CHEMICAL PROPERTIES OF WOOD AND ALLIED SUBSTANCES

(Compiled from Various Government Publications)

Wood	Weight per Cu. Ft. Air Dried Lb.	Weight per Cu. Ft. Green Lb.	Gross Heat Value B.t.u. per Lb. (Kiln-Dried)	Ultimate Analysis, Per Cent (Dry) Carbon	Hydrogen	Oxygen	Ash
Ash, white	12	47	8210	49.73	6.93	43.04	0.30
Beech	13	54	8063	51.64	6.26	41.45	0.65
Birch, white	38	51	7958	49.77	6.49	43.45	0.29
Cedar, white	21	28	7725	48.80	6.37	44.46	0.37
Cypress	29	47	9078	54.98	6.54	38.08	0.40
Elm	44	53	8105	50.35	6.57	42.34	0.74
Fir	27	52	8285	52.32	6.42	41.23	0.03
Hemlock	25	19	8000	52.38	5.91	41.23	0.48
Hickory, shellbark	57	65	7980	49.67	6.49	43.12	0.73
Maple	44	58	8414	51.55	6.61	41.28	0.56
Oak, black	42	61	7530	48.78	6.09	44.98	0.15
red	45	65	7988	49.49	6.62	43.74	0.15
white	48	59	8112	50.44	6.59	42.73	0.24
Pine, pitch	36	54	10120	59.00	7.19	32.68	1.12
white	27	39	8176	52.55	6.08	41.25	0.12
yellow	29	49	8836	52.60	7.02	40.07	0.31
Poplar	29	49	8211	51.64	6.26	41.45	0.65
Corn (air dried)	56 lb. per bushel		8160				
Straw (dry, compressed) (white)	6-8 lb. per cu. ft.		6500				
Tanbark			9500	51.80	6.04	40.74	1.42

Kiln-dried wood has a moisture content of approximately 8 per cent; Air dried, about 12 to 15 per cent, and green wood 25 to 60 per cent.

Charcoal is made from wood in much the same manner that coke is made from coal. It is seldom used for steam generation except in plants where it is a waste by-product.

Tanbark is the fibrous portion of bark remaining after its use in the tanning industry. The ultimate analysis of dry tanbark is practically the same as that of the wood from which it is taken, and its heating value in the dry state is about 9500 B.t.u. per lb. Tanbark, when removed from the vats, is very wet (moisture content about 65 per cent), and it is usually fed to the furnace in this condition. The net heat available for boiler service is very low because of the excessive moisture content, and is approximately 2700 B.t.u. per lb. Tanbark is an unimportant fuel because of its limited use.

Tanbark as a Fuel: Trans. A.S.M.E., Vol. 29, 1909; Vol. 30, 1910.

Bagasse, or Megasse, as it is sometimes called, is refuse sugar cane and is used as fuel on the sugar plantations. The chief constituents are (1) fiber, (2) sucrose, glucose and other reducing sugars, and (3) water. The fiber content varies from 50 to 60 per cent of the total weight; the sucrose and other reducing sugars from an almost negligible quantity to 10 per cent; and the water from 40 to 65 per cent. When bagasse is fired in the raw state, the gross heating value varies from 3600 to 4800 B.t.u. per lb., depending upon the moisture content. In the dry state, the heating value is approximately 8300 B.t.u. per lb. Bagasse is burned either in the raw state or after being wholly or partially dried. One ton of Louisiana sugar cane generates from 1.16 to 1.44 b.hp. It is thus seen, considering the thousands of tons of sugar cane raised, that bagasse is an important fuel in the sugar house.

Bagasse as a Fuel: A.S.M.E., Vol. 39, 1917, p. 611.
The Heat Value of Corn Power: Aug 8, 1922, p. 211.

20. Clinkering and Non-Clinkering Fuels. — Clinker is formed by the mechanical adhesion of the particles of ash or by the fusion of the ash itself. From the operating standpoint, the clinkering characteristics of a fuel are of greater importance than all others, with the possible exception of the caking, or so-called "coking," properties. The standard ash-fusion temperature is taken as 2450 deg. fahr., with a variation of 50 degrees plus or minus. If the ash-fusion temperatures are below 2400 deg. fahr., the fuels are classified as **clinkering,** and if above 2500 degrees as **non-clinkering. Hard clinker** is formed by the direct melting of the ash or of some of its constituents. It hardens while in the ash on the grates. **Soft clinker** remains molten while on the grates, but hardens when its temperature is sufficiently reduced. All solid fuels containing ash will clinker when the rate of combustion is sufficiently high, but whether the resulting clinker is objectionable or not can be determined only by actual service test. Large amounts of non-adhering clinker are not particularly objectionable, while small amounts of pasty slag may give much trouble. There appears to be no definite relation between the chemical composition of the ash and its clinkering properties, because of the influence of such factors as construction of the furnace, combustion space, draft, cooling action of the grates, and the like. As a rule, ash that is high in silica contains little iron and will not fuse easily; but if the silica decreases and the iron increases, fusing will take place at a lower temperature. The curves in Fig. 9 give some idea of the relation of fusing temperature of ash to the percentages of silica, iron oxide, and sulphur. The softening or fusing temperature, as determined in the laboratory, is a measure of the clinkering quality of the fuel, as ash that gives a fusing temperature above 2700 deg. fahr. will rarely give trouble if the

coal is properly fired. In the anthracite coals and the lower and older bituminous beds, the ash is refractory (2600 to 3100 deg. fahr.), giving no trouble from fusion. The bulk of the Pennsylvania bituminous beds have medium ash fusibility (2200 to 2600 deg. fahr.). In the central and western region, the ash fuses readily (1900 to 2200 deg. fahr.). Clinkering fuels give the best results when handled on stokers that clear themselves of ash continuously. With coking coals, the fuel bed should be agitated during combustion; with free-burning coal it should not be disturbed.

21. Calorific Power of Solid Fuels.—The heat liberated by the complete combustion of a unit weight of fuel is called the **heating value,** or **calorific power,** of the fuel. The only accurate method of determining this quantity for a solid fuel is to burn a weighed sample in an atmosphere of oxygen in a suitable calorimeter. An alternative method is to calculate the heating value from the ultimate analysis. Approximate results may be obtained from empirical formulas based upon the proximate analysis.

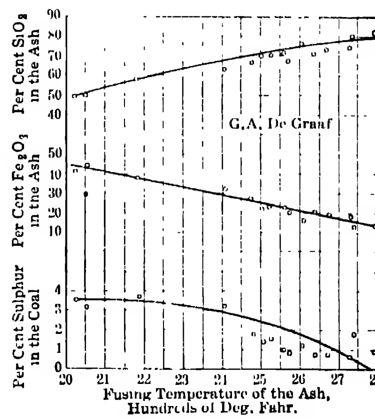

FIG. 9. Curves showing Relation of Fusing Temperature of Ash to Percentage Content of Silica, Iron Oxide, and Sulphur.

Dulong's formula is the generally accepted rule for calculating the heating value of coal. It is based on the assumption that all the oxygen in the fuel, and enough hydrogen to unite with it, are inert in the form of water, and that the remainder of the hydrogen and all of the carbon and sulphur are available for oxidation, thus:

$$h_d = 14{,}600\ C + 62{,}000\ (H - O/8) + 4000\ S, \text{[1]} \tag{4}$$

in which h_d = heating value in B.t.u. per lb. of fuel.

C, H, O and S refer to the proportion by weight of carbon, hydrogen, oxygen, and sulphur in the fuel.

Heating values calculated by means of Dulong's formula fail to check with calorimetric determinations, because

(1) The heating values of the elements, carbon, hydrogen, and sulphur, are not accurately established and the true values may depart somewhat from those given in the formula.

[1] In the fuel bulletins of the U. S. Geological Survey and the Bureau of Mines, Dulong's formula is stated:

$$h_d = 14{,}544\ C + 62{,}028\ (H - O/8) + 4050\ S.$$

(2) The heating value of an element in a chemical compound is not necessarily that of the same element in the free state, because of absorption or evolution of heat during formation of the compound.

(3) The oxygen content in the ultimate analysis is determined by difference. This method throws the summation of all the errors incurred in the other determinations upon the oxygen. Furthermore, the assumption that all of the oxygen is combined with hydrogen to form water is not true, since some of the oxygen may be combined with carbon.

However, in spite of these objections, extensive investigations show that Dulong's formula gives results which agree substantially with calorimetric determinations for all ordinary coals. With lignite, wood, and other fuels high in oxygen, and with some fuels high in hydrogen, such as cannel coal, the results are not reliable and may be considerably in error.

Numerous attempts have been made to establish empirical formulas for calculating the heat value from the proximate analysis, but the results have been decidedly discordant. Many of these rules give consistent results when applied to certain classes of fuels or to fuels from a given district, but as general laws they may lead to serious error.

In this connection may be mentioned the investigations of Mahler,[1] Lord and Haas,[2] Parr and Wheeler,[3] Goutal,[4] and Kent.[5]

When a series of tests is being made with a view of improving efficiency, it is of considerable importance to have the results of each test immediately after completion of the run, in order that the information gained may be used in the succeeding tests. For this reason it is particularly desirable to determine the heating value of the coal and "cinders" with as little delay as possible. If the source of the coal supply is known, the simplest, and a fairly accurate method, is to assume a fixed heat value for the combustible. This may be obtained from results of previous tests or from results published by the Bureau of Mines. For example, the average heat value of the combustible for a number of Illinois coals, as compiled from Government reports and other sources, is 14,300 B.t.u. per lb. With the exception of a very few samples, the actual heating value varied less than 2 per cent from this average and the maximum departure did not exceed 3 per cent. Extensive experiments conducted in the power plant laboratory of Armour & Company, Chicago, Illinois, show that the heat value of the combustible in the refuse or clinkers is

[1] Steam Boiler Economy, R. T. Kent. John Wiley & Sons, Inc., 1915, p. 143.

[2] Trans. A.S.M.E., Vol. 27, 1897, p. 259.

[3] Illinois University Engineering Experiment Station, Bul. 37, 1909.

[4] Comptes Rendus de L'Academie des Sciences, Vol. 135, p. 477.

[5] Trans. A.S.M.E., Vol. 36, 1914, p.189.

practically that of the combustible in the fuel, averaging 14,100 B.t.u. per lb. for Illinois coals.

The heating value of any fuel may be determined from the proximate analysis, with a fair degree of accuracy, by calculating the ultimate analysis, as shown in the preceding paragraphs, and applying Dulong's formula.

Calorimetric determinations are necessary in all cases where accuracy is required.

Example 5. — Approximate the heat values for the Illinois coal (analysis as in Example 1) from the calculated ultimate analysis.

Solution. — Proceed as in tabular chart.

	B.t.u. per Lb. of Coal as Received	Departure from Calorimeter Determinations Per Cent
1. Assuming a fixed heat value for the combustible $h = 14{,}300 \times 0.8163$	11,674	−2.36
2. Calculated from Dulong's formula:		
* (a) $h = 14{,}600 \times 0.65 + 62{,}000 \times 0.0326 + 4000 \times 0.028$	11,623	−1.96
† (b) $h = 14{,}600 \times 0.682 + 62{,}000 (0.0435 - 0.0775/8)$	12,053	+0.80
‡ (c) $h = 14{,}600 \times 0.6655 + 62{,}000 (0.0428 - 0.0751/8) + 4000 \times 0.0197$	11,869	−0.76
3. Actual value from calorimeter test	11,957	0.00

* (a) Ultimate analysis calculated from average analysis of Illinois coals. See Example 4.
† (b) Ultimate analysis calculated from proximate analysis (Equations (1) to (3)).
‡ (c) Ultimate analysis from chemical tests.

22. Size of Coal. — Coal is marketed in different sizes, varying from lump to screenings. The latter furnish by far the greater part of the stoker fuel used. The sizes and grades vary so much, according to kind and locality, that there are no generally recognized standards. The standards recommended by the A.S.T.M. and A.S.M.E. are given in Tables 4 and 5.

Specific Gravity Studies of Illinois Coal, Univ. of Ill., Bul. No. 44, July 3, 1916.
Weight of Various Coals: Bureau of Mines, Tech. Paper, No. 184, 1918.

For maximum efficiency, coal should be uniform in size. With hand-fired furnaces there is usually no limit to its fineness, and larger sizes can be used than with stokers. As a rule, the percentage of ash increases as the size of coal decreases. This is due to the fact that all of the fine foreign matter separated from larger coal, or which comes from the roof or the floor of the mine, naturally finds its way into the smaller coal. The size best adapted for a given case is dependent upon the intensity of draft, kind of stoker or grate, and the method of firing, and its proper

TABLE 4

ANTHRACITE COAL SIZES

A.S.T.M. Standard

Trade Name	Diam. of Opening Through or Over Which Coal Will Pass, Inches		Trade Name	Diam. of Opening Through or Over which Coal Will Pass, Inches	
	Through	Over		Through	Over
Broken .	$4\frac{1}{2}$	$3\frac{1}{4}$	Buckwheat #1 (Buckwheat)	$\frac{1}{2}$	$\frac{1}{4}$
Egg .	$3\frac{1}{4}$	$2\frac{3}{8}$	Buckwheat #2 (Rice)	$\frac{1}{4}$	$\frac{3}{16}$
Stove .	$2\frac{3}{8}$	$1\frac{1}{4}$	Buckwheat #3 (Barley) . .	$\frac{3}{16}$	$\frac{3}{32}$, $\frac{1}{16}$
Chestnut	$1\frac{1}{4}$	$\frac{3}{4}$	Culm	$\frac{3}{32}$, $\frac{1}{16}$	
Pea . . .	$\frac{1}{4}$	$\frac{1}{2}$			

TABLE 5

BITUMINOUS COAL SIZES

A.S.M.E. Standard

Trade Name	Diam. of Opening Through or Over which Coal Will Pass, Inches		Trade Name	Diam. of Opening Through or Over which Coal Will Pass, Inches	
	Through	Over		Through	Over
Eastern Coals					
Run of Mine	As Mined		Nut .	$1\frac{1}{4}$	$\frac{3}{4}$
Lump . .		$1\frac{1}{4}$	Slack	$\frac{3}{4}$	. . . .
Western Coals					
Run of Mine	As Mined		Nut 3-in.	3	$1\frac{1}{4}$
Lump, 6-in.		6	Nut $1\frac{1}{4}$-in. . .	$1\frac{1}{4}$	$\frac{3}{4}$
Lump, 3-in.		3	Nut $\frac{3}{4}$-in. .	$\frac{3}{4}$	$\frac{5}{8}$
Lump, $1\frac{1}{4}$-in.		$1\frac{1}{4}$	Screenings	$1\frac{1}{4}$	
Franklin Co., Ill. Standard					
Small egg (#1)	3	2	Pea (#4)	$\frac{3}{4}$	$\frac{1}{4}$
Stove (#2) .	2	$1\frac{1}{4}$	Carbon (#5)	$\frac{1}{4}$	
Chestnut (#3) .	$1\frac{1}{4}$	$\frac{3}{4}$			

selection often affords an opportunity to effect considerable economy. The influence of the size of screenings on the capacity and efficiency of a boiler in a specific case is illustrated in Fig. 10. The curves are plotted from a series of tests conducted with Illinois screenings on a 500-hp. B. & W. boiler, equipped with chain grates, at the power house of the Commonwealth Edison Company. For sizes of washed coal see paragraph 23.

Selective Preparation of Boiler Fuel Combustion, Feb. 1923, p. 98.

23. Washed Coal. — Coal is washed for the purpose of separating from it such impurities as slate, sulphur, bone coal, and ash. All of these impurities show themselves in the ash when the coal is burned. Screenings contain anywhere from 5 per cent to 25 per cent of ash and from 1 per cent to 4 per cent of sulphur. Washing eliminates about 50 per cent of the ash and some of the sulphur. The evaporative power of the combustible is practically unaffected by washing, and the greater part of the water taken up by the coal is removed by thorough drainage. Many coals, otherwise worthless as steam coals, are rendered marketable by washing. There is no recognized standard of sizes for washed coal, the sizes and grades varying according to kind and locality. The following sizes apply to Williamson County only, but give some idea of the average dimensions for other localities:

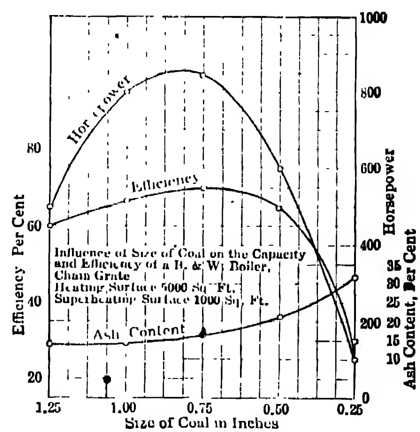

Fig. 10 Influence of Size of Coal on Boiler Capacity and Efficiency.

	Through	Over
No. 1	3 -in. round holes	$1\frac{1}{4}$-in. round holes
2	$1\frac{3}{4}$-in. " "	1 -in. " "
3	1 -in. " "	$\frac{3}{4}$-in. " "
4	$\frac{3}{4}$-in. " "	$\frac{1}{4}$-in. " "
5	$\frac{1}{4}$-in. " "	

Coal Washing in Illinois: Univ. of Ill., Bul. No. 9, Oct. 27, 1913.

24. Selection and Purchase of Coal. — Perhaps no single item in the operation of an existing plant, or in the design of a new plant, affords such an opportunity for effecting economy as the selection of fuel. Care-

ful investigations have shown that almost any fuel can be efficiently burned in suitably designed special furnaces; therefore the problem of selecting a fuel for a proposed installation requires experience with the different kinds of equipment, in addition to a thorough knowledge of the characteristics of various fuels. For existing plants, the problem is largely a matter of testing. In many cases it has been found advisable to redesign furnaces to utilize a low-grade fuel rather than to purchase an expensive coal. The following information is useful in deciding on the coal best adapted for a plant:[1]

a. Type and size of boilers and furnaces.
b. Load conditions, average and maximum loads.
c. Draft available and method of control.
d. Character of coals offered or available.
  1. Moisture and its effect on weight of combustible.
  2. Volatile matter and its relation to type of furnace.
  3. Ash; its amount and its fusibility and tendency to clinker.
  4. Sulphur; the amounts and how combined.
  5. Heating value, calorimeter determination.
  6. Coking qualities of the coal.
  7. Storage and tendency to spontaneous combustion.
e. Relation of the size of coal to the equipment.

After the desired grade of fuel has been decided upon, the next step is to enter into an agreement with the dealer whereby the delivery of that particular fuel may be depended upon. The important items to be considered in the specifications are:

a. A statement of the amount and character of the coal desired.
b. Conditions for delivery.
c. Disposition to be made of the coal in case it is outside the limits specified.
d. Correction in price for variation in heating value and in moisture and ash content.
e. Method of sampling.
f. By whom analyses are to be made.

In specifying the character of the coal desired for the average small plant, every essential requirement of the purchaser may be fulfilled by confining the specifications to the four following characteristics:

Moisture,
Ash,
Size of coal,
Calorific power of coal.

[1] *The Purchase of Coal,* Dwight, T. Randall. Trans. A.S.M.E., Vol. 31, 1911, p. 987.

Although moisture is a great and uncertain variable, and the producer can exercise no control over this factor, the purchaser should protect himself against excessive moisture by stipulating an amount consistent with the average inherent moisture in the coal, and proper penalty should be fixed for delivery in excess of the amount allowed, a corresponding bonus being paid for delivery of less than contract amount. Considerable attention should be given to the percentage of earthy matter contained. The amount of earthy matter usually fixes the heating value of the coal, since the heating value of the combustible is practically constant. The effect of ash on the fuel value of Illinois screenings, as fired under a B. & W. boiler with chain grate, is shown in Fig. 11. This value varies with the different types of boilers, grates, and furnaces, but is substantially as illustrated. The amount of refuse in the ashpit is always in excess of the earthy matter as reported by analysis, except where the amount carried beyond the bridgewall is very large.

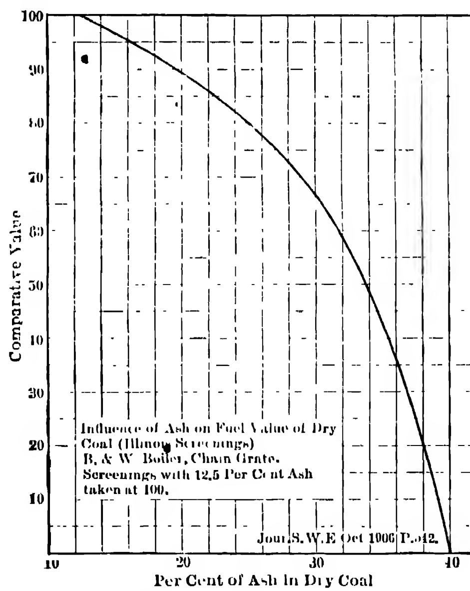

FIG. 11 Influence of Ash on Fuel Value of Dry Coal.

The maximum allowable amount of sulphur is sometimes specified, since some grades of coal that are high in sulphur cause considerable clinkering. Sulphur, however, is not always an indication of a clinker-producing ash, and a more rational procedure would be to classify a coal as clinkering or non-clinkering according to its behavior in the particular furnace in question, irrespective of the amount of sulphur present. An analysis of the various constituents of the ash is necessary to determine whether or not the sulphur unites with them to produce a fusible slag; and as such analyses are usually out of the question on account of the expense attached, they may well be omitted. Ash fuses between 2000 and 3000 deg. fahr., and if the formation of objectionable clinker is to be avoided the furnace must be operated at temperatures below the fusing temperature. Several large concerns insert an "ash fusibility" clause in their coal specifications.

The heating value of the coal, as determined by a sample burned in an

atmosphere of oxygen, does not give its commercial evaporative power, since this depends largely upon the composition of the fuel, character of grate, and conditions of operation. It serves, however, as a basis upon which to determine the efficiency of the furnace. In large plants where a number of grades of fuel are available, it is customary to conduct a series of tests with the different grades and sizes, and the one that evaporates the most water for a given sum of money, other conditions permitting, is the one usually contracted for. In designing a new plant, particular attention should be paid to the performance of similar plants already in operation, and the fuel and stoker that are found to give the best returns for the money should be the ones selected. Where smoke prevention is a necessity, the smoke factor greatly influences the choice of fuel and stoker.

A Rational Basis for Coal Purchase Specifications, E. B. Ricketts, Proc. Am. Soc. Testing Mat., Vol. 22, 1922, p. 557

Effective B.t.u. and Cost Determine Value of Coal, Power, Sept. 18, 1923, p. 448.

25. Powdered Fuels. — Practically all solid fuels can be burned efficiently when finely ground or pulverized. In fact, some of the overall boiler and furnace efficiencies realized with powdered fuels have been equal to those obtained in the best oil-burning plants and 2 to 3 per cent higher than those of the best stoker-fired plants. The problem, however, of whether a fuel should be burned in bulk or powdered form is largely a financial one, in which increased heat efficiency must be balanced against the ultimate cost of obtaining this efficiency. Each system of firing has its advantages and disadvantages, and what may be of small consequence in one situation may prove a serious drawback in another, so that all the items entering into the problem must be carefully studied before an intelligent choice can be made. Numerous cases may be cited in which anthracite, all grades of bituminous, lignite, and peat are giving excellent results when burned in the powdered form; but the art has not yet been developed to the point where sufficient data are available to prove conclusively that the same results could not have been obtained with properly installed and operated stoker-fired plants. In view of the latest developments, it seems probable that within a few years powdered fuel will supplant the mechanical stoker in certain fields, while in others the stoker may extend its service. With low-grade bituminous coals, anthracite culm, lignite, and peat, dust firing appears to have the advantage; but with a good grade of bituminous coal, anthracite, and coke breeze, the stoker-fired plant is still the better investment, except, perhaps, where the load factor is very low and the standby losses correspondingly high. Some of the advantages obtained in burning powdered fuel are as follows:

(*a*) *Complete combustion may be obtained.* The fuel, in the form of fine impalpable dust, is forced or induced into the zone of combustion, where each minute particle is brought into contact with the necessary amount of air, and complete oxidation is effected with minimum air excess.

(*b*) *Overall heat efficiency is increased.* Heat efficiencies of boiler, furnace, and grate, as high as 85 per cent have been obtained on test with stoker-fired boilers; but with normal operation, considering all standby losses, overall efficiencies seldom exceed 78 per cent, and this only in the very best practice where highly skilled help is employed. With a correctly designed and properly operated powdered fuel system, efficiencies as high as 87 per cent have been obtained on test, and overall efficiencies as high as 80 per cent have been maintained on continuous operation. With feedwater economizers and air preheaters, efficiencies as high as 93 per cent have been recorded.

(*c*) *A cheaper grade of fuel may be burned.* In fact, some grades of fuel which are burned with only moderate success in bulk may be efficiently consumed in the powdered form. With stokers and hand-firing, the boiler, furnace, and grate efficiency drops off with the decrease in heat value of the fuel, but such is not the case with the powdered product. Powdered fuel practically eliminates loss of combustible in the ashpit.

(*d*) *The fuel and air supply may be readily controlled to meet the fluctuations in load.* A heavy overload can be quickly taken on, or dropped, without the waste of fuel that frequently occurs under like conditions in stoker practice. During banked periods no fuel is fired. Both the stack damper and auxiliary air inlets may be closed tightly; hence no air flows through the furnace. The standby losses are reduced to a minimum.

The factors which must be considered in connection with powdered fuel, and which may affect the problem of selection are:

(1) *First cost of fuel-preparation plant.* There is no question but that the first cost of the equipment from " coal car to ash car " is greater for plant using the powdered fuel than for the stoker-fired plant, but the difference in cost depends upon so many conditions that general figures are of little value.

(2) *Size of plant.* The minimum size of boiler plant which can be operated more economically as a pulverized-fuel plant than as a stoker-fired plant depends upon whether the fuel is prepared in a central plant or in the so-called " unit " plant. (See paragraph 113.) For the central plant this minimum has been placed as low as 500 and as high as 3000 b.hp. rated capacity. Unit plants as small as 100 hp. are purported to give economical returns on the investment. The market price of the equipment, cost of fuel and labor, and the load factor of the plant are the

controlling elements. The largest powdered-coal-burning plant to date is that of the Cahokia station of the Union Electric Light & Power Company, which is to have an ultimate capacity of 300,000 kw.

(3) *Space requirements.* The extra space required to take care of the fuel-preparation plant may prove an obstacle to the installation of such a plant; but a study of the latest powdered-coal central stations will show that in a modern boiler room, specially designed for powdered fuel, the entire apparatus may be compactly housed.

(4) *Cost of preparing powdered fuel.* The cost of preparing powdered fuel, exclusive of fixed charges, depends largely upon the initial condition of the fuel, desired fineness of the powdered product, size of plant, cost of fuel and labor, and the quantity of fuel handled. In plants under 2500 b.hp. rated capacity, the operating cost of firing fuel is generally greater for the powdered equipment than for a modern stoker installation, but in larger plants the cost is approximately the same. The average cost of preparing powdered coal in a number of plants of 9000 to 2500 rated b.hp. (year 1923) ranged from 35 to 60 cts. per ton; this covers the entire cost, including fixed charges, from the unloading of the coal to its delivery in the furnace.

Cost of Preparing and Delivering Powdered Coal to the Furnace: Bureau of Mines, Bul. 217, 1923, p. 100, National Engineer, Nov. 1924, p 528.

(5) *Maintenance.* As most of the pulverizing-plant equipment is of the slow-moving type, the maintenance cost may be kept to practically that of the coal- and ash-handling equipment of a stoker-fired plant of equivalent size. Much trouble has been experienced because of the rapid destruction of furnace brickwork in improperly designed powdered-fuel furnaces, but the latest installations show that the maintenance cost of the refractories is no greater than with stoker firing.

(6) *Storage.* Large quantities of powdered fuel cannot be stored economically for any great length of time, because of its hygroscopic properties and its tendency to pack when moist. Many cities limit the storage of powdered fuel to such small quantities as to interfere seriously with operation in case of breakdown to the pulverizing or drying system. In the modern powdered-fuel plant, sufficient reserve capacity is effected by intermediate storage between furnace and mill.

(7) *Slagging.* At high boiler ratings, with fuels having a low fusing point, considerable slagging of the ash occurs. This molten slag is very destructive to the brickwork with which it comes in contact. The same objections, however, hold true for stoker-fired plants. In the latest installations no trouble is experienced from slagging.

(8) *Ash discharge into the atmosphere.* From 10 to 30 per cent of the

ash content of the fuel may be discharged through the stack into the atmosphere. The material discharged is very fine and flocculent, and the greater portion of it remains suspended in the air until precipitated by moisture. In the Middletown Plant of the Metropolitan Edison Company the ash remaining in the flue gas is removed by cinder-vane induced-draft fans, while in the Trenton Channel Plant of the Detroit Edison Company it is precipitated electrostatically.

(9) *Overload capacity.* Boilers equipped with powdered fuel furnaces have not yet reached the extreme overload capacity of underfeed stoker-fired installations, but that is merely a question of design and not an inherent limitation.

See paragraph 115 for powdered-coal furnaces and paragraph 126 for a description of powdered-coal handling systems.

Wide Range of Fuels Possible through Pulverization. Combustion, July, 1923, p. 26.
Pulverized Fuels in Central Station Boiler Rooms. Combustion, July, 1923, p. 26.
Pulverized Fuel. Report of Prime Movers Committee, N.E.L.A., Sept., 1925.

LIQUID FUELS

26. General. — Any combustible liquid may be burned efficiently in a properly designed furnace. Liquid fuels offer many advantages over solid fuels from the operating standpoint, as will be shown later, but, with the exception of mineral oils, they are usually too costly for steam generation in stationary plants. Vegetable and animal oils, and even alcohol, have been burned commercially in power plant furnaces, but only under unusual conditions. Mineral oil, or petroleum, furnishes by far the greater part of the liquid fuel used for power and heat generation. Approximately 50 per cent of the annual production of mineral oil is available as fuel, but improved processes of "cracking" are resulting in larger yields of gasoline and the lighter distillates, so that the percentage of residue or fuel oil is becoming less as the art of cracking progresses. Oil has been recognized for years as the paramount fuel of marine service, and particularly of navy requirements, and so vital has its use become in this direction that it plays an important part in the policies of nations, and is a matter of international concern. Of course, there is always a possibility that new fields may be opened up, or that oil may be economically produced from oil-shale lignite, or coal, or from industrial by-products; but it is doubtful if the ultimate quantity will ever be great enough, or cheap enough, to compete seriously with coal for stationary plants within coal-producing zones. Oil will be used where it is the most commercially efficient source of heat and power, because of absence or inadequate supply of cheaper fuels, and where the use of oil as fuel represents an economical means of disposing of excess accumulations of crude oil,

residue, or distillates. The use of fuel oil for domestic heating and other low-pressure steam generators is rapidly increasing, but the factors to be considered in this connection differ widely from those affecting the large power house.

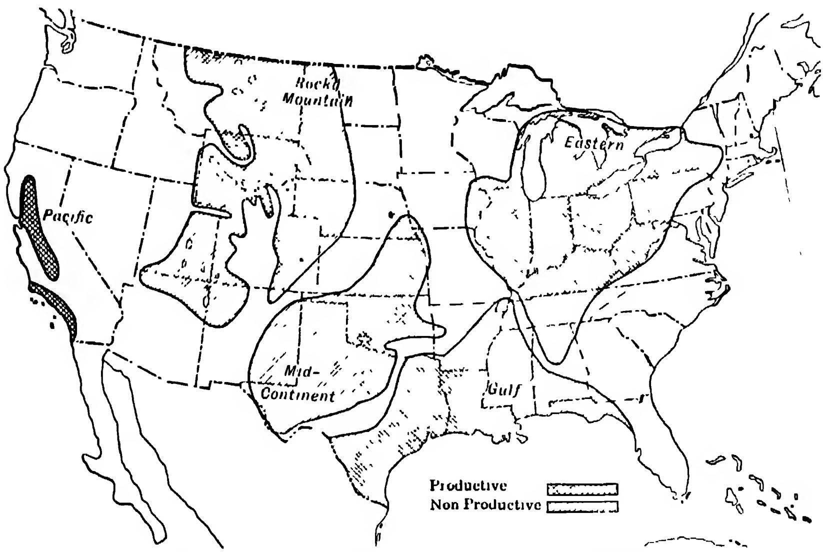

FIG. 11*a* Oil-field Groups of the United States.

27. Source of Oil-fuel Supply. — Most of the oil fuel consumed in the United States is a by-product of the manufacture of gasoline from crude oil, though some of the lower grades of crude oil are burned as mined, with only a small amount of "topping." The greater part of the supply

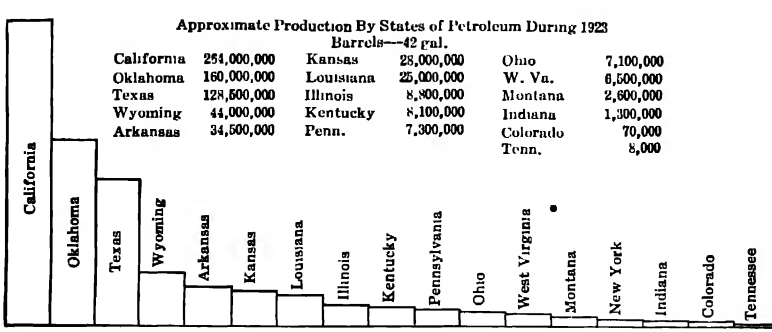

FIG. 12. Estimated Production of Fuel Oil During 1923 in the United States.

is of domestic production, notwithstanding the large quantity imported from Mexico. A rough indication of local availability of oil fuels to consumers is found in the monthly reports of the U. S. Bureau of Mines, in-

cluding stocks on hand at refineries. Many centers of distribution, however, are remote from refineries. Tar and tar-oil obtained as a by-product from gas works, petroleum distilleries, and coke ovens, are excellent fuels and may be burned in much the same manner as fuel oils. Because of the valuable "coal-tar" products obtainable from the crude tar, it is questionable if the cost will be low enough to permit its use as a boiler fuel, except possibly in the immediate vicinity of the producing plants.

28. Physical and Chemical Properties of Oil Fuels. — Petroleum, or crude oil, as pumped from the wells, consists principally of carbon and hydrogen, together with small amounts of oxygen, sulphur, nitrogen, water in emulsion, and silt. The oxygen and nitrogen may be classified with the moisture and silt as inert impurities. The sulphur, though combustible, has a low calorific value and is otherwise undesirable. It is common practice to divide crude oils into three general groups, as (1) paraffin-base, (2) intermediate-base and (3) asphalt-base. This classification is one that is not always applied accurately, nor are authorities agreed as to what properties are characteristic of the several groups. E. W. Dean, Petroleum Chemist, of the U. S. Bureau of Mines, believes that the determining factor which should be accepted is the relative content of hydrocarbons of two distinct chemical series. On this basis it may be stated that paraffin-base crudes are those containing relatively high percentages of aliphatic hydrocarbons (the aliphatic series includes the so-called paraffins) and low percentages of cyclic hydrocarbons (the naphthenes are cyclic hydrocarbons). Naphthene-base crudes contain relatively high percentages of cyclic and low percentages of aliphatic hydrocarbons. Intermediate-base crudes are, as the name indicates, intermediate in properties between the two extreme classes.

The most clearly defined property that serves to differentiate the classes is that of gravity. Paraffin-base products of a given boiling range have low specific gravities (high Baumé gravities), whereas naphthene-base distillates of the same volatility have high specific gravities or low Baumé gravities. The property of viscosity also serves as a distinguishing quality, as paraffin-base oils have lower viscosities than naphthene-base products of the same volatility. This is in line with the generally recognized fact that naphthene-base oils have lower flash points than paraffin-base oils of the same viscosity. From Table 6, it will be seen that the ultimate chemical constituents of the various grades and classes of mineral oils vary but slightly, while the physical properties vary widely. For example, the crude oils analyzed in the Table differ greatly in volatility, specific gravity, and viscosity, but have approximately the same percentages of carbon and hydrogen.

TABLE 6

PHYSICAL AND CHEMICAL PROPERTIES OF TYPICAL OIL FUELS

Arranged According to Baumé Gravity

Kind of Fuel	Physical Properties				Chemical Properties				
	Gravity		Flash Point, Deg Fahr Open Cup	Viscosity Saybolt Seconds at 70 Deg Fahr	C	H	O+N	S	B.t.u. per Lb.
	Baumé at 60 Deg. Fahr.	Specific 60/60							
Crude oils									
Western field..	14.37	0.970	192	387	85.64	11.37	0.84	1.06	18,478
Do	16.34	0.957	172	380	86.58	11.61	0.74	0.82	18,613
Do	17.52	0.950	230	250	86.37	11.30	1.14	0.60	18,727
Mid-continent.	20.00	0.933	273	220	86.06	11.21	2.06	0.67	19,440
Do	22.17	0.920	180	198	84.60	10.90	2.87	1.63	19,060
Do	24.00	0.909	264	188	87.93	11.37	0.19	0.41	19,650
Eastern	31.70	0.866	102	165	86.40	12.37	1.02	0.13	19,890
Do	40.10	0.823	78	112	82.00	14.80	3.20	0.10	20,300
Fuel oils			Closed Cup						
Mexican ...	10.00	1.000	374	322*	88.05	7.58	1.00	3.28	17,500
Do ...	11.82	0.987	310	300*	86.90	9.36	..	3.16	18,050
Do ...	14.20	0.971	208	92†	85.26	10.31	.	3.94	18,260
Mid-continent	16.60	0.955	190	74†	83.22	10.16	3.74	2.83	18,427
Western	18.00	0.946	186	750‡	86.11	11.81	1.20	0.67	18,790
Mid-continent	22.10	0.920	181	389‡	83.26	12.41	3.83	0.59	19,430
Do	24.35	0.907	179	350‡	85.40	13.07	1.12	0.15	19,235
Do	28.26	0.884	175	290‡	81.90	13.70	1.40	0.37	19,312
Do	32.00	0.861	160	76‡	84.82	13.09	2.05	0.36	19,511
Do	35.70	0.845	154	42‡	84.36	13.21	1.09	0.32	19,627
Gas oil.	16.10	0.958	195	98†	86.16	9.05	3.06	0.30	16,960
Oil tar ..	..	1.150	485	418†	92.00	6.13	0.22	0.33	17,300
Coal tar	.	1.250	530	490†	89.20	4.95	5.27	0.56	15,800
Kerosene, 150 deg.	47.00	0.791	110	...	85.16	14.33	0.51	...	19,922

* At 250 deg fahr. † At 220 deg. fahr ‡ At 100 deg. fahr

Very little crude is burned as mined. The average fuel oils in the Middle West are those which are classified according to their Baumé gravity as 24/26, 26/28, 28/30 and 30/32. The Mexican fuel oils on the Gulf and Atlantic Coast are practically all 10/12 and 14/16 gravity. The ultimate analyses (per cent by weight) of these oils are substantially as follows:

	28/36 Baumé	22/28 Baumé	14/22 Baumé	10/12 Baumé
Carbon ..	84.0	85.0	86.0	87.0
Hydrogen ...	13.0	12.0	11.0	9.5
Sulphur	0.3	0.5	0.8*	1.1*
Nitrogen...	0.2	0.2	0.2	0.2
Oxygen........	1.0	1.0	1.0	1.0
Water....	0.2	0.5	1.0	1.2

* The sulphur content of some fuel oils, notably those from Mexico, is as high as 4 per cent.

The "Flash," "Fire," and viscosities are approximately as follows:

Gravity	Flash	Fire	Viscosity, Saybolt Seconds at 100 Deg. Fahr.	B.t.u. per Lb.
10/12	400	430	320*	18,000
14/16	300	325	100*	18,430
24/26	185	220	350	19,290
26/28	170	205	305	19,330
28/30	165	190	215	19,410
32/36	155	175	41	19,610

* 250 deg. fahr

TABLE 7

RELATION BETWEEN BAUMÉ GRAVITY AND WEIGHT PER BARREL AND PER GALLON

Degrees Baumé	Specific Gravity 60°/60°	Weight per Barrel	Pounds, per Gallon	Degrees Baumé	Specific Gravity 60°/60°	Weight, per Barrel	Pounds, per Gallon
10	1.0000	350.0	8.33	25	0.9032	316.1	7.52
11	0.9929	347.5	8.27	26	0.8974	314.1	7.47
12	0.9859	345.1	8.21	27	0.8917	312.1	7.42
13	0.9790	342.7	8.16	28	0.8861	310.1	7.38
14	0.9732	340.3	8.10	29	0.8805	308.2	7.33
15	0.9655	337.9	8.04	30	0.8750	306.2	7.29
16	0.9589	335.6	7.99	31	0.8696	304.3	7.24
17	0.9524	333.4	7.93	32	0.8642	302.5	7.20
18	0.9459	331.1	7.88	33	0.8589	300.6	7.15
19	0.9396	328.9	7.83	34	0.8537	298.8	7.11
20	0.9333	326.7	7.78	35	0.8485	297.0	7.07
21	0.9272	324.5	7.72	36	0.8434	295.2	7.02
22	0.9211	322.4	7.67	37	0.8383	293.4	6.98
23	0.9150	320.3	7.62	38	0.8333	291.6	6.94
24	0.9091	318.2	7.57	39	0.8284	289.9	6.90

TABLE 8

COMPARATIVE HEAT VALUES OF SOLID FUELS AND FUEL OIL

Heat Value of Solid Fuel B.t.u. per Lb.	Lb. of Solid Fuel Equal to One Barrel of Oil		Barrel of Oil Equal to 1 Ton (2000 Lb.) of Solid Fuel	
	B.t.u. per Lb. 18,500*	B.t.u. per Lb. 19,500†	B.t.u. per Lb. 18,500*	B.t.u. per Lb. 19,500†
6000	1048	984	1.91	2.07
7000	900	843	2.23	2.37
8000	787	738	2.54	2.71
9000	700	656	2.86	3.05
10000	629	590	3.18	3.38
11000	572	536	3.50	3.73
12000	524	492	3.82	4.14
13000	484	454	4.13	4.41
14000	448	421	4.45	4.75
15000	419	393	4.77	5.08

About 16 deg. Baumé. † About 32 deg. Baumé.

Fuel oil is usually measured in terms of barrels of 42 gals., at a standard temperature of 60 deg. fahr. One barrel of oil weighs from 310 to 350 lb., according to the specific gravity. Compared with coal, oil occupies about 50 per cent less space and is approximately 35 per cent less in weight for equal heat value. For rough estimation, the coefficient of expansion of fuel oil may be taken as 1/10 for every 40 deg. fahr. For exact values, see Density and Thermal Expansion of American Petroleum Oils, Circular No. 57, and Technologic Paper No. 77, U. S. Bureau of Standards.

Present Status of Oil Fuel: Combustion, Jan., 1923, p. 32.
Manual for Oil and Gas Operation: Bul. 232, 1923 Bureau of Mines.

29. Calorific Power of Oil Fuels. — The true heating value of liquid fuels can be found only by direct calorimeter measurements. For rough approximations, Dulong's formula may be used, but this requires a knowledge of the ultimate constituents of the fuel. Empirical rules based on specific gravity appear to give fairly consistent results, but no one rule is applicable to all liquid fuels. For California anhydrous crude oils, Prof. J. N. Le Conte gives the following:

$$\text{B.t.u. per lb.} = 17{,}680 + 60\text{ B} \tag{5}$$

in which B = degrees Baumé at 60 deg. fahr.

Another rule given by Sherman and Kropff, purporting to be applicable within an error of 2 per cent to all liquid petroleum products, is:

$$\text{B.t.u. per lb.} = 18{,}650 + 40\ (\text{B} - 10) \tag{6}$$

Other things being equal, oils rich in hydrogen have a higher calorific value per pound than those rich in carbon, but a lower value per gallon. A barrel of heavy crude oil will, therefore, have a higher heat value than a barrel of lighter oil. For general comparisons, the heat values of coal and fuel oil are substantially as indicated in Table 8.

For standard methods of testing oil, see Chapter XVII.

30. Advantages and Disadvantages of Liquid Fuels for Steam Generation. — Since mineral fuel oil constitutes the greater part of the fuel burned in boiler furnaces, the following statements refer specifically to this class of fuel:

(1) Pound for pound, the heat value of oil is approximately 35 per cent higher than that of high-grade coal.

(2) For equal heat values, the space required for the storage of oil is about 50 per cent less than that for coal.

(3) The burning of oil causes no dust or ash; there is no cleaning of fires.

(4) There is no loss in heat value due to deterioration while in storage.

(5) Stack losses are low, because the air excess required for complete combustion is reduced to a minimum.

(6) There is greater adaptability to variation in load; automatic regulation of fires is readily effected.

(7) Standby losses are reduced to a minimum.

(8) Boiler room labor is less than with coal firing, owing to elimination of coal- and ash-handling.

(9) High combustion efficiency may be attained.

Some of the disadvantages are as follows:

(*a*) Insurance liability is usually higher than with solid fuels. Civic and insurance requirements may impose burdensome or prohibitive restrictions on the location and arrangement of the storage tanks, etc.

(*b*) Maintenance cost of furnace refractories is high, unless furnace is specially designed for oil burning.

(*c*) The degree of superheat is less with oil than with coal, for a given equipment and load.

(*d*) There is an element of uncertainty as to the delivery of large quantities, and extreme fluctuation in cost.

(*e*) Nearly all fuel oil burners of the steam atomizing type produce an objectionable roaring sound.

The real criterion in the selection of fuel is the ultimate cost of producing energy in the required form; and since this depends on countless factors which vary with each proposed installation, general deductions are without purpose.

Furnaces and Equipment for Burning Liquid Fuels: See paragraphs 116, 117, and 127.

Practical Uses of Fuel Oil: Combustion, Feb., 1923, p. 94.
Coal Tar as a Source of Fuel: Gas Age-Record, July 28, 1923.

31. Colloidal Fuels. — Colloidal fuel is a name given to an emulsion of powdered solid fuel and oil, which was developed in this country by the Submarine Defense Association to meet war conditions. A so-called fixateur is used to stabilize the elements of the mixture into a homogeneous product. Most oils in their natural state may be mixed with pulverized anthracite, lignite, peat, coke, or wood, to produce smokeless colloidal fuel. It is possible to produce a stable liquid containing 40 per cent of the powdered product. The colloidal fuel is fired with the same equipment as fuel oil, and with approximately the same overall efficiency. The heat value of the composite fuel is naturally dependent upon the heat value and weight of the constituent fuels. The heat value per unit volume is

greater than that of straight oil, unless the powdered component has a very low heat value and specific gravity. For example, in a composite made up of 35 per cent by weight of powdered anthracite (14,000 B.t.u. per lb., sp. gr. 1.6) and 65 per cent of oil (18,200 B.t.u. per lb. sp. gr. 0.96) the calorific value will be 165,000 B.t.u. per gallon against 146,000 B.t.u. for the oil. The use of colloidal fuels will effect a large saving in oil. should conditions arise which would make this a commercially economical procedure. Owing to its solid fuel content, colloidal fuel is heavier than water and may, therefore, be stored under a water seal. At this date very little has been done with colloidal fuels in connection with stationary steam plants.

Tests of Colloidal Fuel Power, April 29, 1919, p 662.
Plastic Fuel or "Amalgam" Power, Dec 27, 1921, p. 1032.

GASEOUS FUELS

32. General. — Gaseous fuels, on account of their simple molecular structure, can be burned readily and without smoke in any commercial apparatus from a boiler furnace to a gas engine. Such fuels are in the ideal form for perfect combustion, and permit of simple automatic control. They have all the advantages of liquid and solid fuels, with none of the disadvantages, save that they are not sufficiently concentrated for convenient storage. Unfortunately, gaseous fuels are prohibitive in cost for steam generation, except when the plant is favorably located with respect to natural gas wells or when the gaseous fuel is a by-product from some industrial process. Because of the heat losses in conversion from solid or liquid to gaseous form, and because of the plant investment that is necessary, there is no economy in manufacturing gas solely for steam generation. The most commonly used gaseous fuels for steam generation are natural gas, blast-furnace gas, and by-product coke-oven gas.

Gas-fired Boilers. Combustion, Feb., 1924, p. 110.

33. Natural Gas. — The demand for natural gas for purposes other than steam generation is steadily increasing, so that even in the immediate vicinity of the wells the cost is frequently higher than that of other classes of fuel. Natural gas is composed primarily of carbon and hydrogen in varying proportions, with small quantities of nitrogen, oxygen, and occasionally sulphur. The gaseous constituents, H_nC_m, vary within a wide range, and it is practically impossible to give an average analysis which means anything. For example, some gases are practically all methane, CH_4, while others are extremely high in C_2H_6 or C_2H_2. Practically all natural gases contain some CO, and a number of them contain

CO_2. The heat value ranges from 720 to 1700 B.t.u. per standard cu. ft. with an average of about 1100.

Composition of Natural Gas: Tech Paper No. 109, 1915, Bureau of Mines.
Analysis of Natural Gas: Tech. Paper No. 104, 1915, Bureau of Mines.
Liquefied Products from Natural Gas: Tech. Paper No. 10, 1912, Bureau of Mines.

34. Blast-furnace Gas. — As the name implies, blast-furnace gas is a by-product from the blast furnace of the iron industry. Coke furnishes about 90 per cent of the fuel used in this connection, and its consumption per ton of pig iron varies from 1600 to 3600 lb., with an average of 2000. The weight of gas produced per ton of pig iron varies according to the weight of coke, gaseous constituents of the flux and coke, weight of oxygen combined with the material charged, and the weight of air delivered by the blowing engine. For rough approximations, it is satisfactory to allow 70 cu. ft. of gas per lb. of coke. The heat value of the gas varies from 85 to 110 B.t.u. per cu. ft. under standard conditions and ranges in composition approximately as given in Table 9. Blast-furnace gas, as it leaves the furnace, is very dirty, each cu. ft. containing as much as 227 grains of dust in suspension. The dust content is reduced by suitable means before it is fed to the furnace.

Burning Blast-furnace Gas Under Boilers Power, Dec. 13, 1921, p. 930.
Combustibility of Blast-furnace Gas. The Blast Furnace and Steel Plant, Aug., 1922.

TABLE 9

PROPERTIES OF TYPICAL FUEL GASES

Gas	Constituents of Gas — Per cent by Volume								Calorific Value B.t.u. per Standard Cu. Ft.*	
	CO_2	CO	H_2	CH_4	C_2H_4	H_nC_m	N_2	O_2	High	Low
Blast-furnace	11.4	28.6	2.7	0.2	..	..	57.1	..	102	100
do ..	10.9	27.8	2.8	0.2	..	..	58.3	..	96	94
Coke-oven ...	2.5	6.0	42	34.3	2.0	2.0*a*	10.1	1.1	605	546
do	0.8	4.9	54.2	28.4	..	..	10.1	1.6	479	426
Illuminating:										
Coal ..	1.2	6.2	43.9	37.8	5.9	4.2*a*	3.5	0.5	618	558
Water .	4.4	44.8	45.6	4.4	0.1	..	0.1	0.5	336	311
Water carbureted	2.1	24.1	32.4	23.4	12.5	..	3.7	0.5	638	596
Natural .	6.5	..	..	84.3	8.0	..	1.2	..	987	900
do	0.3	0.6	1.2	93.6	0.2	..	3.4	0.6	940	860
do	0.2	0.5	1.9	92.8	0.2	..	3.8	0.4	992	900
do ..	0.4	..	35.9	49.6	12.3	0.6*b*	..	0.8	836	750
do	0.0	..	..	32.3	..	67.0*b*	0.7	..	1420	1350
Oil..........	0.9	..	24.3	58.3	17.4	..	..	..	940	758
do	..	0.2	4.8	53.7	41.2	..	0.1	..	1220	1125
Producer:										
Anthracite ...	5.3	26.1	15.0	0.2	.	..	53.2	0.2	135	127
Bituminous ..	3.4	25.3	9.2	3.1	0.8	..	58.2	..	155	146
Lignite	10.6	14.1	13.8	2.6	0.4	..	58.2	0.3	110	101
Peat..... ..	12.1	27.2	0.9	3.1	0.1	..	56.7	.	122	118

* 68 deg. fahr.: 14.7 lb. per sq. in. *a*. C_6H_6. *b*. C_2H_6.

35. By-product Coke-oven Gas. — This gas is a by-product in the manufacture of coke by the destructive distillation of coal. Instead of burning the gaseous distillate at its point of origin, as in a beehive or retort coke oven, it is conducted through suitable apparatus and cooled, yielding tar, ammonia, illuminating and fuel gas. A certain portion of the gas is burned in the oven, and the remainder is available for fuel or illuminating purposes. By-product gas is ordinarily saturated with moisture and carries a large proportion of tar and hydrocarbon. The heat value of the gas varies from 400 to 550 B.t.u. per standard cu. ft. and varies in composition approximately as given in Table 9.

36. Calorific Power of Gases. — The heating value of a combustible gas may be accurately determined by means of a "flow" calorimeter, such as the Junker and Boyce. The heating value thus obtained is the higher or absolute value and the only one to be used in connection with steam boiler practice. The heating value may be calculated, with sufficient accuracy for all commercial purposes, by assuming each constituent gas to be free and uninfluenced by the others; thus, if a gas is composed of 80 per cent by volume of CH_4 and 20 per cent CO, the heating value per cu. ft. of the mixture will be 0.8 of the heat value of CH_4, B.t.u. per cu. ft., + 0.2 of the heat value of CO, B.t.u. per cu. ft. This method, of course, requires a knowledge of the character, amount, and heating value of the gaseous constituents. The standard cu. ft. (A.S.M.E. Code) is taken at 68 deg. fahr. and 29.92 in. of mercury (14.7 lb. per sq. in.). Another standard frequently used by gas manufacturers is based on a temperature of 62 deg. fahr. The conversion from a volume to a weight basis (and *vice versa*) at any pressure and temperature is readily made by means of equation (8).

Example 6. — Calculate the heat value of by-product coke-oven gas having the following analysis by volume:

Per Cent by Volume.
CO_2 0.8; O_2 1.6; CO 4.9; CH_4 28.4; H_2 54.2; N_2 10.1.

Solution. — The CO_2, O_2 and N_2 have no heating value; hence, they need not be considered. The heating value of each gas may be taken from Table 11.

Constituent	Cu Ft.	B.t.u. per Standard Cu Ft of Constituent	Calculation	
CO	0 049	318	0 049 × 318	16.0
CH_4	0 284	992	0.284 × 992	282 0
H_2	0 542	325	0 542 × 325	176 0
			Total B.t.u. per cu. ft.	474.0

37. Town Refuse and Garbage. — The composition of unsorted refuse from different cities varies within wide limits, but in a general sense is approximately one-third each of combustible matter, ash, and water, and the heat value of the combustible is roughly one seventh that of coal. From a fuel standpoint, the heat value is too low to compensate for the capital outlay on the destructor plant; but as refuse destruction by burning is fundamentally a sanitary measure, it may prove economical to utilize part of the heat of combustion as a by-product for power generation. • A description of the various destructors used in this connection is beyond the scope of this work, and the reader is referred to the accompanying references for further study.

Garbage and Refuse Disposal Data: Municipal Journal, Vol. 26, 1918, p. 318; Municipal Engineering, Sept., 1919 p. 107; Engr. News Record, October 17, 1918, p. 715; Trans. A.S.M.E., Vol. 39, 1917, p. 689, 779; Journal Western Society Engineers, Vol. 22, 1917, p. 623.

PROBLEMS

1. The following analyses were obtained from samples of Illinois coal "as received."

Proximate Analysis		Ultimate Analysis			
	Per Cent		Per Cent		Per Cent
Moisture.	8.10	Hydrogen	5.1	Oxygen	15.2
Volatile matter..	32.50	Carbon	62.5	Sulphur ..	3.5
Fixed carbon.....	46.80	Nitrogen	1.1	Ash ..	12.6
Ash.....	12.60				
	100.00			Total ...	100.00

a. Transfer these analyses to the "moisture-free" and "moisture and ash free" basis.
b. Transfer the ultimate analysis to the "moisture, ash, and sulphur free" basis.
c. Determine the free hydrogen, combined moisture, and total moisture.
d. Calculate the ultimate analysis "as received" from the ultimate analysis.

2. If the moisture, sulphur, and ash content of an Illinois coal "as fired," are 12.6, 3.5 and 8.1 per cent, respectively, estimate the ultimate analysis by Evans' method (see Example 4).

3. Calculate the heat value of the coal "as fired," "moisture free," and "moisture and ash free," analyses as in Example 1.

4. Compare the following fuels on a "B.t.u. for one cent" basis: Wood — $4.00 per cord, assuming 65 per cent interstitial space, 40 lb. per cu. ft. and 6000 B.t.u. per lb.; anthracite — $9.00 per ton of 2000 lb., 12,500 B.t.u. per lb.; pocahontas — $7.50 per ton, 14,000 B.t.u. per lb.; bituminous nut — $5.50 per ton, 13,000 B.t.u. per lb.; bituminous slack — $4.00 per ton, 10,000 B.t.u. per lb.; gas — 80 cents per 1000 cu. ft., 600 B.t.u. per cu. ft.; 26 degree Baumé fuel oil — $2.00 per bbl., 19,300 B.t.u. per lb.

CHAPTER III

COMBUSTION OF FUELS

38. Elementary Theory. — So far as the engineer or fireman is concerned, the theory embodied in the combustion of fuels is very elementary and involves the simplest of mathematics; but it should be pointed out that a complete analysis of the complicated phenomena involved in gas reactions requires a knowledge of chemical equilibrium and kindred subjects that lies beyond the scope of this book.

To the chemist, combustion is the phenomenon resulting from any chemical combination evolving heat. To the engineer, it means the chemical union of the combustible of a fuel and the oxygen of the air at such a rate as to cause rapid increase of temperature. Such combinations always occur in accordance with fixed and immutable laws, both as regards weight relationship and volume changes. The union liberates a definite quantity of heat, directly proportional to the mass of material taking part in the reaction, and independent of the time occupied. The heat thus generated when a unit weight of substance is completely burned is called the **heating value,** or **calorific power,** of that substance. Before chemical union can take place, the combining elements must first be brought to the **ignition temperature** or **kindling point** of the combustible. The temperature necessary to cause this union of oxygen and combustible has been fairly well established for simple combustible gases, but there is no definite temperature at which complex fuels, such as coal and allied substances, burst into flame. Experiments show that coal liberates heat of combustion at all temperatures. The point at which the coal assumes a uniform glow has been taken as the most logical ignition temperature.[1] The ignition temperature of a number of gases and the glow point of several solid fuels are given in Table 10. These values are approximate only, since the temperature may vary with the relative amount of surface of the substance, pressure of the air, and the presence of other substances that aid reactions, but they serve for the purpose at hand. Matter is never destroyed; hence, all of the elements composing a fuel and its air requirements will be found in the products of combustion after the fuel has been "burned," but, barring certain inert elements, they will be in different combinations with each other. The reactions taking place during combustion are generally expressed by simple molecular equations

[1] *The Ignition Temperature of Coal,* Bul. No. 128, Apr. 10, 1922, Univ. of Ill. Engineering Exp. Station.

in which the elements are designated by symbols, the relative volumes of the gaseous constituents by numerical coefficients, and the number of times the atomic weight occurs by subscripts. The symbols, relative atomic and molecular or combining weights, and the chemical reactions for the elements and compounds generally encountered in combustion work are given in Table 11.

Flame Temperatures, by Prof. W Trinks, Power, June, 1923.

Combustion Phenomena, by E Kieft, Combustion, Nov., 1923, p. 390.

TABLE 10

IGNITION TEMPERATURE, DEG. FAHR.

Acetylene	900	Ethylene, CH	1020
* Anthracite Coal	1112	Hydrogen	1130
* Bituminous Coal	850	* Lignite	979
* Coke	1123	Methane CH	1200
Carbon Monoxide	1200	* Semi-Bit. Coal.	980
Ethane CH	1000	Sulphur	470

* Glow point.

TABLE 11

DATA RELATIVE TO ELEMENTS MOST COMMONLY MET WITH IN CONNECTION WITH COMBUSTION OF FUELS

Substance	Chemical Symbol	Relative Combining Weight (O_2 = 32) Exact	Approx	Chemical Reaction	Heating Value B t u. per Lb Higher	Lower
Acetylene	C_2H_2	26.03	26	$2\,C_2H_2 + 5O_2 = 4\,CO_2 + 2\,H_2O$	21,600	21,000
Carbon to CO_2	C	12.005	12	$C + O_2 = CO_2$	14,600	14,600
Carbon to CO	C	12.005	12	$2\,C + O_2 = 2\,CO$	4,440	4,440
Carbon monoxide	CO	28.01	28	$2\,CO + O_2 = 2\,CO_2$	4,354	4,354
Ethane	C_2H_6	30.05	30	$2\,C_2H_6 + 7\,O_2 = 4\,CO_2 + 6\,H_2O$	22,230	20,500
Ethylene	C_2H_4	28.03	28	$C_2H_4 + 3\,O_2 = 2\,CO_2 + 2\,H_2O$	21,600	20,420
Hydrogen	H_2	2.015	2	$2\,H_2 + O_2 = 2\,H_2O$	62,100	52,920
Methane	CH_4	16.03	16	$CH_4 + 2\,O_2 = CO_2 + 2\,H_2O$	23,850	21,670
Sulphur to SO_2	S	32.06	32	$S + O_2 = SO_2$	4,000	4,000
Sulphur to SO_3	S	32.06	32	$2\,S + 3\,O_2 = 2\,SO_3$	5,940	5,940

Gas	Chemical Symbol	Relative Combining Weight O_2 = 32 Exact	Approx	Density and Volume* Lb per 100 Cu Ft	Cu Ft per Lb.	Air Required for Perf Combus † Lb per Lb of Gas	Cu Ft per Cu Ft of Gas	Heating Value per Cu. Ft * Higher	Lower
Acetylene	C_2H_2	26.03	26	6.76	14.79	13.35	11.90	1460	1420
Carbon monoxide	CO	28.01	28	7.27	13.75	2.48	2.38	318	318
Ethane	C_2H_6	30.05	30	7.82	12.78	16.16	16.70	1735	1600
Ethylene	C_2H_4	28.03	28	7.30	13.70	14.85	14.30	1573	1491
Hydrogen	H_2	2.016	2	0.52	192.0	34.80	2.38	325	278
Methane	CH_4	16.03	16	4.16	24.00	17.32	9.52	992	902
Air	‡	28.95	29	7.52	13.30				
Carbon dioxide	CO_2	44.00	44	11.43	8.75				
Nitrogen	N_2	28.02	28	7.28	13.74				
Oxygen	O_2	32.00	32	8.31	12.03				
Sulphur dioxide	SO_2	64.07	64	16.65	6.00				

* 68 deg. fahr. and atmospheric pressure. † See paragraph 44. ‡ Equivalent to O_2 + 3.82 N_2.

39. Combustion of Carbon. — Carbon is the principal combustible of nearly all fuels. The primary product of the oxidation of carbon is a complex of carbon and oxygen, which has a transitory existence. This complex decomposes into a mixture of **carbon dioxide** (CO_2) and **carbon monoxide** (CO) in proportions dependent upon the temperatures at which decomposition take place. If CO_2 is the ultimate product resulting from the combustion of pure carbon, combustion is said to be " perfect "; if CO is formed it is said to be " incomplete," because the monoxide is in itself combustible and capable of further oxidation; if both CO_2 and free O_2 are present combustion is said to be complete.

When carbon and oxygen unite to form CO_2 the reaction is expressed by:

$$C + O_2 = CO_2 \quad (7)$$

[1]

The combining weights involved are: C, 12; O_2, 32; CO_2, 44. Introducing these, we have

$$12 + 32 = 44$$

Divided by 12,

$$1 + 2\tfrac{2}{3} = 3\tfrac{2}{3}$$

Thus, 1 lb. of carbon unites with $2\frac{2}{3}$ lb. of oxygen to form $3\frac{2}{3}$ lb. of carbon dioxide. If the CO_2 resulting from this combustion is cooled at constant pressure to the initial temperature of the original mixture of carbon and oxygen, the heat liberated will be about 14,600 B.t.u. per lb. of carbon. (The heat value for carbon appears to depend upon the method of preparation and ranges according to various authorities from 14,220 to 14,647 B.t.u. per lb.)

The oxygen required for combustion is usually taken from the atmosphere. For most engineering purposes, dry atmospheric air may be taken as a mechanical mixture of oxygen and nitrogen in the ratio 23 to 77 by weight, and 21 to 79 by volume. For convenience in calculation, these ratios may be expressed as follows:

Nitrogen = 77/23 × oxygen = 3.34 × oxygen, by weight.
Air = 100/23 × oxygen = 4.35 × oxygen, by weight.
Nitrogen = 79/21 × oxygen = 3.76 × oxygen, by volume.
Air = 100/21 × oxygen = 4.76 × oxygen, by volume.

Nitrogen is inert under ordinary furnace conditions and passes into the products of combustion without change. It simply dilutes the oxygen for combustion, and its presence in the flue gases represents a large per-

[1] The molecular weight of C is not definitely known. Carbon exists in a number of forms, each of which probably has its own molecular weight. Thus the difficulty of burning carbon in the form of soot is attributed to its complex molecular structure.

centage of the heat discharged to waste. The presence of nitrogen in the ordinary furnace, however, is actually an advantage, because it reduces the temperature of combustion below the fusing point of the furnace refractories. If air instead of pure oxygen is used, the minimum theoretical weight of dry air required for the perfect combustion of carbon to CO_2 is therefore $2\ 2/3 \div 23/100 = 11.58$ lb. The weight of nitrogen brought in with the air is $2\ 2/3 \times 77/23 = 8.92$ lb. per lb. of carbon, and the products of combustion will consist of 3.66 lb. of CO_2 and 8.92 lb. of N_2, a total of 12.58 lb.

Gaseous elements, mixtures, and compounds are usually measured volumetrically. The transfer from a weight to a volume basis is readily effected by the following application of **Avogadro's law.**[1]

$$\frac{PV}{T} = \frac{1544}{m} \tag{8}$$

in which

P = pressure, lb. per sq. ft.
V = specific volume, cu. ft. per lb.
T = absolute temperature, deg. fahr.
m = molecular weight of the gas referred to oxygen as 32.

With 68 deg. fahr. and 14.7 lb. per sq. in. as the standard (A.S.M.E. Code), for temperature and pressure, respectively, equation (8) reduces to the convenient form

$$V = 385 \div m \tag{9}$$

Thus, under the assumed standard conditions, the volume of one pound of

Oxygen	$= 385 \div 32 =$	12.03 cu. ft.
Nitrogen	$= 385 \div 28 =$	13.75 cu. ft.
CO_2	$= 385 \div 44 =$	8.75 cu. ft.

Since air is composed of 23 parts by weight of oxygen and 77 parts of nitrogen, its molecular weight is $0.23 \times 32 + 0.77 \times 28 = 28.92$, and its specific volume under standard conditions is $385 \div 28.92 = 13.3$ cu. ft. Strictly speaking, a gaseous mixture cannot have molecular weight, but the number 28.92 in connection with air may be considered the apparent molecular weight.

Referring to the perfect combustion of 1 lb. of carbon with dry air, the volumes (at 68 deg. fahr. and atmospheric pressure) involved in the reaction are

[1] Equal volumes of all gases contain the same number of molecules when at the same temperature and pressure.

Oxygen	$= 2.66 \times 12.03 =$	32.0 cu. ft.
Nitrogen	$= 8.92 \times 13.75 =$	122.5 cu. ft.
CO_2	$= 3.66 \times 8.75 =$	32.0 cu. ft.
Air	$= 11.58 \times 13.3 =$	154.0 cu. ft.

It will be seen that the volume of CO_2 is precisely the same as that of the oxygen used in the process, and since oxygen forms 21 parts of air by volume, it follows that with perfect combustion the products will consist of 21 per cent of CO_2 and 79 per cent of nitrogen. The same conclusion may be reached in a simpler manner, by noting the fact that numerical coefficients in the molecular equations represent relative volumes. Considering the coefficients in equation (7) we have:

$$1 \quad 1 = 1$$

which signifies that 1 volume of oxygen combines with carbon to produce 1 volume of CO_2. That is, the volume of CO_2 resulting from combustion is exactly the same as that of the oxygen supplied, both gases referred to the same pressure and temperature.

When combustion is incomplete and the carbon unites with oxygen to form CO, the reaction is expressed:

$$\begin{aligned} 2C + O_2 &= 2CO\,^{1} \\ 2 \times 12 + 32 &= 2 \times 28 \\ 1 + 1\,1/3 &= 2\,1/3 \end{aligned} \qquad (10)$$

Thus, 1 lb. of carbon combines with 1 1/3 lb. of oxygen to produce 2 1/3 lb. of CO. The heat liberated will be about 4410 B.t.u. per lb. of carbon. The air required to furnish the necessary oxygen for this reaction is 1 1/3 $\div$ 0.23 = 5.79 lb. The weight of nitrogen brought in with the air is 1 1/3 $\times$ 77/23 = 4.46 lb., and the products of combustion will consist of 2.33 lb. of CO and 4.46 lb. of nitrogen, a total of 6.79 lb.

Considering the coefficients in equation (10), we have

$$2 \quad 1 = 2$$

which indicates that 1 volume of oxygen combines with carbon to produce 2 volumes of CO. The reaction is therefore accompanied by an increase in volume of 100 per cent.

[1] All of the carbon does not burn directly to CO. Part of it burns first to CO_2, and this in turn combines with carbon to form CO. The ultimate result, however, is the same as if the change took place as indicated in this equation.

Carbon monoxide unites with oxygen to form CO_2, thus

$$\begin{aligned} 2\,CO + O_2 &= 2\,CO_2 \qquad (11)\\ 2 \times 28 + 32 &= 2 \times 44 \\ 56 + 32 &= 88 \\ 1 + 4/7 &= 1\,4/7 \\ 2\,1/3 + 1\,1/3 &= 3\,2/3 \end{aligned}$$

That is, 1 lb. of CO unites with 4/7 lb. of oxygen to form 1 4/7 lb. of CO_2; or, the 2 1/3 lb. of CO resulting from the combustion of 1 lb. of carbon, as in equation (10), combines with 1 1/3 lb. of oxygen to form 3 2/3 lb. of CO_2. The heat liberated will be about 4354 B.t.u. per lb. of CO, or 2 1/3 × 4354 = 10,160 B.t.u. per lb. of carbon. Noting that 4440 + 10,160 = 14,600, it is evident that the ultimate result is the same whether the process takes place in one or two stages. The dry air required to furnish the necessary oxygen for the combustion of 1 lb. of CO to CO_2 is 4/7 ÷ 0.23 = 2.47 lb. The weight of nitrogen brought in with the air is 4/7 × 77/23 = 1.91 lb. per lb. of carbon, and the products of combustion will consist of 0.57 (= 4/7) lb. of CO_2 and 1.91 lb. of N_2, a total of 2.48 lb. The fact that carbon may combine with oxygen to form CO_2, CO, or both, is of great importance in furnace efficiency and is discussed at greater length in paragraph 54.

40. Combustion of Hydrogen. — Hydrogen combines with oxygen to form water vapor, thus:

$$\begin{aligned} 2\,H_2 + O_2 &= 2\,H_2O \qquad (12)\\ 2 \times 2 + 32 &= 2 \times 18 \\ 4 + 32 &= 36 \\ 1 + 8 &= 9 \end{aligned}$$

That is, 1 lb. of hydrogen combines with 8 lb. of oxygen to form 9 lb. of water. If the water vapor resulting from this combustion is all condensed and cooled at constant pressure to liquid at the initial temperature of the original mixture of hydrogen and oxygen, the heat liberated will be about 62,000 B.t.u. per lb. of hydrogen. If, however, the vapor is not condensed, or is only partially condensed, the heat available for external heating will be less than 62,000 B.t.u. by an amount depending upon the final conditions of the products of combustion. These two values are called, respectively, the **higher** and the **lower heat value.** The lower factor is a variable, and therefore cannot have a constant value except for a fixed set of conditions.

The theoretical weight of dry air necessary to burn 1 lb. of hydrogen to water is 8 ÷ 0.23 = 34.8 lb., and the final products of combustion will consist of 9 lb. of H_2O, and 8 × 77/23 = 26.8 lb. of N_2, a total of 35.8 lb.

It is important to note that the water vapor content is never determined in the ordinary gas analysis apparatus, since the greater part is condensed before reaching the measuring pipette. Therefore, the dry products of combustion will consist only of N_2. The lower heat value for the complete combustion of hydrogen, with theoretical air requirements, under constant pressure of 14.7 lb. per sq. in. and initial and final temperature of 62 deg. fahr., is given by Goodenough as 52,930 B.t.u. per lb. of hydrogen.[1] The lower heat value does not enter into boiler problems, since the heat of the water formed by combustion, and discharged with the flue gas, is considered as a separate loss. The A.S.M.E. Power Test Code recommends the use of the higher heat value only.

Oxygen is a constituent of practically all fuels and may exist in the free state, in combination with nitrogen, in organic nitrates, in the carbonates of the ash, in the combined moisture, and in the $C_mH_nO_z$ compounds. For most engineering purposes, it is sufficiently accurate to assume that all oxygen present is combined with hydrogen as H_2O, and that the remainder of the hydrogen is in the free state. With this assumption, considering that 8 parts of oxygen combine with 1 of hydrogen, we have as the **free** or **available hydrogen,** $H - O/8$. The oxygen to be supplied from the air then becomes $8 (H - O/8)$ lb. per lb. of hydrogen, and the weight of the dry air itself, $34.8 (H - O/8)$ lb. per lb. of hydrogen.

41. Combustion of Hydrocarbons. — In nearly all fuels, part of the carbon is united with hydrogen in a great variety of combinations known as hydrocarbons, and constituting the so-called volatile matter; and although the ultimate products of combustions are CO_2 and H_2O, the process of decomposition is often very complicated. For example, in the combustion of coal, in hand-fired furnaces, the volatile matter leaves the fuel bed as complex hydrocarbon compounds. Near the surface of the fuel bed, where the oxygen supply is very low, these hydrocarbons are quickly decomposed, by the high furnace temperature, into soot (carbon), H_2 and CO. The formation of the latter is due to the presence of CO_2 and the small supply of air. At a distance of 1 or 2 feet from the surface of the fuel bed, provided air is admitted above the fire, only a very small amount of hydrocarbons can be found in any state, gaseous, liquid, or solid. The solid substance present in the flames is mostly soot with traces of tar. If oxygen is present in sufficient quantity at the time of distillation of the volatile matter, the hydrocarbons will burn directly to CO_2 and H_2O without first decomposing and depositing soot. The oxygen and air requirements for the perfect combustion of the various hydrocarbons are the same as if the constituent elements were in the free state, but the heat of combustion of the compound differs considerably from

[1] Principles of Thermodynamics, 3rd Ed., p. 295.

that of the separate elements. This is explained by the fact that the hydrocarbon is already a chemical compound, and that the heat of combination or of dissociation must be considered. For example, methane (CH_4) unites with oxygen to form H_2O and CO_2, thus

$$\begin{aligned} CH_4 + 2\,O_2 &= CO_2 + 2\,H_2O \\ (12 + 4) + 2 \times 32 &= (12 + 32) + 2\,(2 + 16) \\ 16 + 64 &= 44 + 36 \\ 1 + 4 &= 2\,3/4 + 2\,1/4 \end{aligned} \qquad (13)$$

That is, 1 lb. of CH_4 combines with 4 lb. of oxygen to form 2 3/4 lb. of CO_2 and 2 1/4 lb. of H_2O. The theoretical dry air requirements are 4 ÷ 23/100 = 17.3 lb. per lb. of CH_4. The heat of combustion, as experimentally determined, is 23,850 B.t.u. per lb. as against 26,400 when calculated from the heat combustion of free carbon and free hydrogen. The hydrocarbons most commonly encountered in boiler room practice are outlined in Table 11.

42. Combustion of Sulphur. — Although carbon, hydrogen, and oxygen are generally considered to be the three important elements in coal, sulphur often constitutes a large portion of the coal substance. The sulphur occurs in a variety of forms, organic and inorganic, but very little information is available on the subject. Sulphur unites with oxygen to form sulphur dioxide (SO_2) or sulphur trioxide (SO_3) depending upon the furnace conditions. In most cases SO_2 is formed, thus:

$$\begin{aligned} S + O_2 &= SO_2 \\ 32 + 32 &= 64 \\ 1 + \ 1 &= \ 2 \end{aligned} \qquad (14)$$

That is, 1 lb. of sulphur combines with 1 lb. of oxygen to form 2 lb. of SO_2. If oxygen is obtained from the atmosphere, the theoretical weight of dry air required to completely burn 1 lb. of sulphur is 1 ÷ 23/100 = 4.35 lb., and the final products of combustion will consist of 2 lb. of SO_2 and 1 × 77/23 = 3.35 lb. of nitrogen, a total of 5.35 lb. The heat of combustion of pure sulphur, when burned to SO_2, is about 4000 B.t.u. per lb. Sulphur, however, seldom exists in a fuel as a free element; hence, the assumption that the total sulphur content, as determined from the ultimate analysis, has a heat value of 4000 B.t.u. per lb. may be considerably in error. Attention should be called to the fact that all of the sulphur in coal does not burn to SO_2 and that a large percentage may be found in the ash. No relationship appears to exist between the total amount of sulphur in the fuel and the amount, after combustion, in the flue gases, coke, or ash.

[*] *The Effect of Sulphur on Steam Production*, by T. A. Marsh. Combustion, Feb., 1924, p. 123.

43. Combustion of a Mixture of Elements. — All commercial fuels consist of a mixture of elements existing either in the free state or in combination with each other. The minimum theoretical weight of air required to completely burn any fuel is the same whether the combustible elements are free or combined; but the heat of combustion of a chemical compound may differ considerably from that based on the heat value of its constituent elements, because of the heat absorbed or given up in the creation of the compound. The character and distribution of the products of combustion depend upon the nature of the fuel, the air supply, and the conditions under which combustion took place. In practically all the furnaces, the combustion of solid fuels takes place in two stages: (1) Combustion in the fuel bed, which includes the distillation of volatile matter and partial combustion or gasification of the fixed carbon; and (2) combustion of the gaseous and other combustibles rising from the fuel bed in the combustion space. With liquid fuels, evaporation and gasification precede ignition and combustion, while with gaseous fuels ignition takes place as soon as the fuel and air mixture has reached the proper temperature for chemical union.

The various steps in the combustion of a bed of coal of uniform thickness on a stationary grate are shown in Fig. 13. At the bottom of the

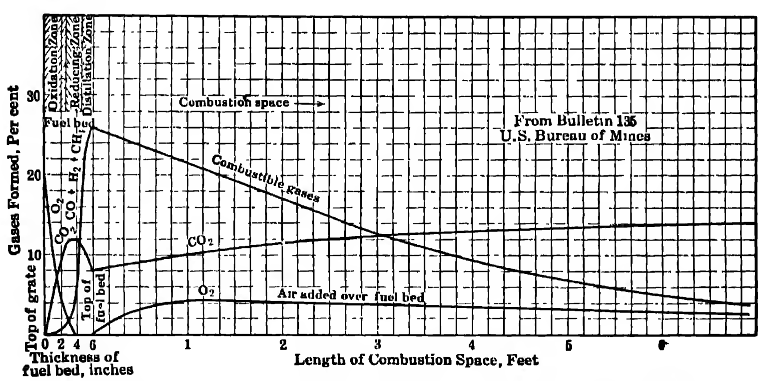

FIG. 13. Composition of Furnace Gases along their Path through the Combustion Space — Hand-fired Furnace.

fuel bed, where the air first comes in contact with the coal, the air contains approximately 21 per cent of oxygen, and the fuel bed but little combustible. As the air passes up through the layer of fuel next the grate, the oxygen in it combines with the carbon of the coal, forming CO_2. The rate of oxidation in the lower part of the fuel bed depends almost entirely on the rate at which air flows through it. The greater the quantity of air that is forced through the fuel bed the faster the coal is oxidized.

When the free oxygen is all used up, the resulting CO_2, on continuing its passage through the superposed unburned portion of the coal bed, is reduced to CO. The rate of reduction of CO_2 to CO depends upon the temperature of the fuel bed — the higher the temperature the faster the CO_2 is reduced to CO. At the high temperature existing in the average fuel bed, a considerable portion of the CO_2 is reduced. The layer at the top of the fuel bed consists mostly of fresh fuel which is being heated and from which the volatile matter is being distilled. With a given temperature, the distillation is independent of the air supply, since the heated volatile matter distills off whether air is supplied or not. On the other hand, fixed carbon in a fuel bed cannot be burned to CO_2 or gasified to CO unless air is supplied through the grate. The gases rising from the fuel bed contain a high percentage of combustible, and no free oxygen, irrespective of the rate at which air is forced through the fuel bed. Therefore, complete combustion cannot be obtained from the air passing through the bed unless there are holes in the fire or part of the fuel on the grate is burned out. To effect complete combustion with an even fuel bed of unburned coal, part of the air must be supplied above the fire and in such a manner that it will mix with the combustible gases. This applies to all solid fuels.

Part of the moisture in a fuel and part of that brought in with the air for combustion pass through the furnace as highly superheated steam. That part of the moisture which comes into contact with the incandescent carbon combines with the carbon to form CO, CO_2, and H_2, thus

$$H_2O + C = CO + H_2$$
$$2\,H_2O + C = CO_2 + 2\,H_2$$

Under average boiler-furnace conditions, the H_2 thus liberated will ultimately recombine with O_2 and form H_2O. See also paragraph 100.

Combustion of Coal: R. B. McMullin, Combustion, Mar., 1922, p. 224 (Serial).

Combustion in the Fuel Bed of Hand-fired Furnaces: Kreisinger, Ovitz and Augustine, Bureau of Mines, Tech. Paper 137, 1917.

Low Rate of Combustion in Fuel Beds of Hand-fired Furnaces: Kreisinger, Augustine and Katz, Bureau of Mines, Tech. Paper 139, 1918.

Experiments with Furnaces for a Hand-fired Tubular Furnace: Flagg, Cook and Woodman, Bureau of Mines, Tech. Paper 34, 1914.

Factors Governing the Combustion of Coal in Boiler Furnaces: Clement, Frazer and Augustine, Bureau of Mines, Tech. Paper 63, 1914.

New Developments in Smokeless Combustion: Power House, Mar. 5, 1923.

44. Air Theoretically Required for Perfect Combustion. — As previously stated, the minimum weight of air required to completely burn any fuel is the same whether the combustible elements are in the free state or in chemical combination with each other. If C, H, O, S, repre-

sent the proportional part by weight of carbon, hydrogen, oxygen, and sulphur in the fuel, then, from the preceding paragraphs, it is evident that for perfect combustion

$$A_0 = 2.66\ C + 8\ (H - O/8) + S \qquad (15)$$

$$A_1 = 11.58\ C + 34.8\ (H - O/8) + 4.35\ S \qquad (16)$$

in which

A_0 = weight of *pure oxygen* per lb. of fuel
A_1 = weight of *dry air* per lb. of fuel

Equation (16) is sometimes expressed as follows:

$$A_1 = 11.58\ C + 34.8\ H + 4.35\ (S - O) \qquad (17)$$

Atmospheric air contains a small amount of water vapor[1] and traces of CO_2, helium, hydrogen, argon, and other elements; but in view of the uncertainty of many of the factors entering into the problem of combustion, it is sufficiently accurate for all engineering purposes to assume that the air is dry and composed only of nitrogen and oxygen in the ratio by weight of 77 to 23.

The constants in equations (15) to (17) will be slightly lower if the exact molecular weights (as fixed by the International Committee on Atomic Weights) are used instead of the approximate weight, and the oxygen-nitrogen ratio of the air is taken as 23.15 to 76.85 instead of 23 to 77. Such refinement, however, is without purpose in engineering practice. The theoretical weight of air per lb. of fuel varies within wide limits, but when expressed in terms of weight per 10,000 B.t.u. there is a close agreement between all solid fuels. Several hundred fuels, varying from peat to anthracite, gave an average value, on this basis, of 7.5 lb. of dry air per 10,000 B.t.u. with a maximum departure not exceeding 4 per cent. See Table 12.

Example 7. — Calculate the minimum weight of dry air necessary to completely burn Illinois bituminous coal having the following analysis:

	Per Cent "as fired"		Per Cent "as fired"
Carbon	65.0	Sulphur	2.8
Hydrogen	4.4	Free moisture	8.8
Oxygen	7.2	Ash	10.5
Nitrogen	1.3	Total	100.0

Solution. — Substituting the value of C, H, O, and S in the equation (16), and solving for A, we have:

$$A = 11.58 \times 0.65 + 34.8\ (0.044 - 0.072/8) + 4.35 \times 0.028$$
$$= 8.87 \text{ lb., the theoretical weight of dry air per lb. of coal as fired.}$$

[1] 0.1 to 1.2 per cent by weight, depending primarily upon the temperature and relative humidity. See paragraph 409.

The "dry coal" is 100 − 8.8 = 91.2 per cent, and the "combustible" 100 − (8.8 + 10.5) = 80.7 per cent of the coal as fired; therefore, the theoretical air requirement is 8.87/0.912 = 9.72 lb. per lb. of **dry coal** and 8.87/0.807 = 10.99 lb. per lb. of **combustible.**

The air requirements for liquid fuels are usually determined by weight, and the method of prodecure is the same as for solid fuels. Gaseous fuels, however, are measured volumetrically and the air supply is calculated on this basis.

Example 8. — Calculate the theoretical air requirements for the perfect combustion of a dry blast-furnace gas having an analysis by volume as follows:

	Per Cent		Per Cent
Carbon dioxide	12.5	Hydrogen	3.5
Carbon monoxide	25.5	Nitrogen	58.5

Solution. — The CO and H_2 are the only combustible elements. From the coefficients of the molecular reactions, equations (11) and (12), it is evident that the CO and H_2 each require one-half their own volume of oxygen for complete combustion or $\frac{1}{2}$ (25.5 + 3.5) = 14.5 cu. ft. per 100 cu. ft. of gas. Since air is composed of 21 per cent by volume of oxygen, we have:

Cu. ft. of air required per 100 cu. ft. of gas = 14.5 ÷ 0.21 = 69 (temperatures and pressures assumed to be constant).

45. Products of Combustion. — The character and amount of the products of combustion resulting from the burning of the unit of any fuel depend upon the nature of the fuel, the air supply, and the conditions under which combustion takes place. For maximum heat efficiency, complete combustion with theoretical air requirements is necessary, and, considering commercial fuels in general, the resulting products should consist only of CO_2, N_2, H_2O, SO_2, ash, and oxides of minor combustible elements. If combustion is complete but air is used in excess of theoretical requirements, the gaseous products will include **free oxygen.** If combustion is incomplete, CO, H_2, soot, and various hydrocarbons may also be present. As the gaseous products resulting from complete combustion are colorless and invisible at chimney temperatures, visible smoke (other than that caused by the condensation of vapor or entrainment of ash particles) is an index to incomplete combustion. If the ultimate analysis of a fuel is available, it is a comparatively simple matter to calculate the character and amount of the products of combustion for complete oxidation; and, *vice versa*, given the character and amount of the products together with the analysis of the fuel, it is possible to determine the amount of air supplied. The calculations are best illustrated by an example.

TABLE 12

THEORETICAL AIR REQUIREMENTS FOR VARIOUS FUELS AND THE RESULTING MAXIMUM PER CENT CO_2 IN THE FLUE GAS FOR PERFECT COMBUSTION

Fuel, Moisture, and Ash Free	Ultimate Analysis					Dry Air, Lb.		CO_2, Per Cent by Volume
	C	H	N	O	S	Per Lb. of Fuel	Per 10,000 B.t.u.	
Pure carbon	100.00	....	....	..	....	11.58	7.8	20.91
Anthracite	94.39	1.77	0.71	2.13	1.00	11.39	7.7	20.06
Semi-anthracite	89.64	3.97	0.63	3.23	2.53	11.59	7.6	20.00
Semi-bituminous	86.39	4.84	1.46	5.50	1.81	11.41	7.6	18.65
Bituminous	79.71	5.52	1.52	9.87	3.38	10.70	7.5	18.46
Sub-bituminous	78.06	5.70	1.35	13.10	1.79	10.24	7.5	18.56
Lignite	70.64	4.61	1.22	22.67	0.86	8.75	7.6	19.68
Peat	59.42	5.50	1.50	33.33	0.25	7.30	7.6	20.79
Crude oil	84.90	13.7	0.60	0.80	.	14.45	7.4	15.90

Example 9. — Determine the character and amount of the products of combustion if 1 lb. of coal, as per following ultimate analysis, is completely burned (1) with theoretical dry air requirements, and (2) with 20 per cent air excess:

	Per Cent as Fired		Per Cent as Fired
Carbon	65.0	Sulphur	2.8
Hydrogen	4.4	Free moisture	8.8
Oxygen	7.2	Ash	10.5
Nitrogen	1.3	Total	100.0

Solution. — The various steps are detailed in the following tabular chart.

Principles of Combustion in The Steam Boiler Furnace, A. D. Pratt Published by The Babcock and Wilcox Co., New York.

Elements of Heat Power Engineering, Hirshfeld and Barnard. Published by John Wiley and Sons, New York.

Combustion in The Power Plant, T. A. Marsh. Published by Combustion Publishing Corp., New York.

Properties of the Products of Combustion, Combustion, Dec. 1923, p. 451.

Reports Relative to Combustion Accessories, Combustion, Aug. 1923, p. 126.

Interpretation of Flue Gas Analysis, Combustion, Feb. 1924, p. 115.

Fuels and Their Combustion, Haslam and Russell. Published by McGraw-Hill Co., New York.

CALCULATED RESULTS

(COAL AS FIRED)

Calculations for Perfect Combustion with Theoretical Air Requirements	Lb. of Substance per 100 Lb. of Coal								
	Flue Gases				Elementary Constituents				
	Dry Gases			H_2O	C	H_2	N_2	O_2	S
	CO_2	SO_2	N_2						
The carbon will produce:									
Carbon					65				
65 × 44/12	238								
65 × 32/12								173	
65 × 32/12 × 77/23			580				580		
The available hydrogen will produce:									
4.4 − 7.2/8						3.5			
(4.4 − 7.2/8)9				31.5					
(4.4 − 7.2/8)8								28	
(4.4 − 7.2/8) 8 × 77/23			94				94		
The oxygen and inert hydrogen will produce:									
Oxygen								7.2	
Hydrogen, 7.2/8						0.9			
Combined H_2O, 7.2 + 7.2/8				8.1					
The sulphur* will produce:									
Sulphur									2.8
2.8 × 64/32		5.6							
2.8 × 32/32								2.8	
2.8 × 32/32 × 77/23			9.3				9.3		
The nitrogen in the fuel is considered inert†			1.3				1.3		
The free moisture will appear as vapor				8.8					
Total	238	5.6	684.6	48.4	65	4.4	684.6	211	2.8

* See end of paragraph (42).

† This is not strictly true, since a portion of the nitrogen content of the fuel appears in the flue gas in combination with other elements, but the amount is so small compared with that supplied in the air that no appreciable error arises from the assumption that it remains inert and passes through the furnace without change.

$$\begin{aligned}\text{Total gaseous products} &= CO_2 + SO_2 + N_2 + H_2O\\ &= 238 + 5.6 + 684.6 + 48.4\\ &= 976.6 \text{ lb. per 100 lb. of coal}\\ &= 9.76 + \text{ lb. per lb. of coal}\end{aligned}$$

Or, separating the compounds into their elementary constituents

$$\begin{aligned}{}^{1}\text{Total gaseous products} &= C + H_2 + O_2 + N_2 + S + \text{Free } H_2O\\ &= 65 + 4.4 + 211 + 684.6 + 2.8 + 8.8\\ &= 976.6 \text{ lb. per 100 lb. of coal}\\ &= 9.76 + \text{ lb. per lb. of coal}\end{aligned}$$

[1] Atmospheric air may contain as much as 1.2 per cent by weight of water vapor; therefore this moisture content should be added if extreme accuracy is desired.

Total *dry* gaseous products = total gaseous products − total H_2O
= 976.6 − 48.4 = 928.2 lb. per 100 lb. of coal
= 9.28 + lb. per lb. of coal

Dry air supplied = total (N_2 + O_2) − (N_2 + O_2) in the fuel
= (684.6 + 211) − (1.3 + 7.2) = 887.1 lb. per 100 lb. of coal
= 8.87 lb. per lb. of coal

(Which checks with the results as calculated from equation (16).)

Considering the second phase of the example; viz., complete combustion with 20 per cent air excess, it will be found that the only change in the products of combustion will be the addition of 20 per cent of the weight of oxygen and nitrogen *furnished by the air*, or 0.20 × 2.038 = 0.407 lb. of free oxygen and 0.20 × 6.833 = 1.367 lb. of nitrogen. Therefore, the total dry gaseous products will consist of 2.38 lb. CO_2; 0.056 lb. SO_2; 6.846 + 1.367 = 8.213 lb. N_2; and 0.407 lb. free O_2, a total of 11.056 lb. per lb. of coal.[1]

In practice, the gaseous products of combustion are measured volumetrically (see paragraph 349), and the various constituents are expressed in per cent of the total volume of dry gas. Equations (8) and (9) offer a simple means for transferring the several quantities from a weight to a volume basis. This is best illustrated by an example.

Example 10. — Calculate the actual volume under standard conditions and the per cent by volume of CO_2, SO_2, O_2, and N_2 in the total dry gaseous products for the data given in the preceding example.

Solution. — Multiply the weight of each constituent by $385/m$, (see equation 9), and the product gives the actual volume under standard conditions. In dealing with ratios by volume, the constant 385 cancels out and need not be considered. Thus, for maximum CO_2,

$$\text{Per cent } CO_2 \text{ by volume} = 100 \frac{2.38 \div 44}{2.38 \div 44 + 6.846 \div 28 + 0.056 \div 64} = 18.1$$

The various steps and results for the other elements are shown in the following tabular chart:

PRODUCTS OF COMBUSTION

(THEORETICAL AIR REQUIREMENTS)

Substance	Weight, Lb. w*	Molecular Weight m	Vol Cu. Ft.		Per Cent by Vol. in Total Dry Gas 100 w/m ÷ 0.299,
			w/m	385 w/m	
CO_2	2.380	44	0.054	20 8	18.1
N_2	6 846	28	0.244	93.9	81.6
SO_2	0 056	64	0 001	0 4	0 3
	9 282		0 299	115 1	100.0

[1] Values have been carried out to three decimal places in order to check with subsequent calculations. In practice such refinement is without purpose since gas analyses are apt to be in error as much as 10 per cent.

PRODUCTS OF COMBUSTION — (*Continued*)
(THEORETICAL AIR REQUIREMENTS)
(Complete combustion with 20 per cent air excess)

CO_2	2.380	44	0.054	20.8	15.0
N_2	8.213	28	0.293	112.8	81.2
O_2	0.407	32	0.013	5.0	3.6
SO_2†	0.056	64	0.001	0.4	0.2
	11.056		0.361	139.0	100.0

* As calculated in Example 9.
† In the commercial analysis of fuel gases, part of the SO_2 is absorbed by the water in the sampling apparatus, while some of it probably goes into the CO_2 pipette. Furthermore, all of the sulphur in the fuel does not burn to SO_2, and a considerable portion may remain in the ash. It is difficult to determine the exact distribution, but since the volume of SO_2 is ordinarily very small, it is common practice to disregard it entirely.

Example 11. — Determine the volume of air necessary per cu. ft. of gas and the resulting per cent of CO_2 in the dry products of combustion, when blast-furnace gas of the following composition by volume is completely burned with theoretical air requirements:

CO_2 11.5 per cent N_2 59 per cent CO 27 per cent H_2 2.5 per cent

Solution. — The CO and H_2 are the only combustible elements. From the molecular reactions, $2\,CO + O_2 = 2\,CO_2$, and $2\,H_2 + O_2 = 2\,H_2O$, it is evident that 2 volumes of CO combine with 1 volume of O_2 to form 2 volumes of CO_2, and similarly, 2 volumes of H_2 combine with 1 volume of O_2 to form 2 volumes of H_2O. That is, 1 cu. ft. of CO and 1 cu. ft. of hydrogen require 1/2 cu. ft. of oxygen each for complete combustion, and 1 cu. ft. of CO will produce 1 cu. ft. of CO_2. Since air is composed of 21 per cent oxygen and 79 per cent nitrogen by volume, the ratio of the nitrogen and of the air to the oxygen is 79/21 = 3.76 and 100/21 = 4.76, respectively, and we may proceed as follows:

Composition of Flue Gas		Calculation	Volume, Cu. Ft.*			
			Theoretical Requirements		Dry Flue Gas	
Element	Cu. Ft.		O_2	Air	CO_2	N_2
CO_2	0.115	Remains unchanged			0.115	
N_2	0.590	" "				0.590
CO	0.270	0.270 × 1.0			0.270	
		0.270 × 0.5	0.1350			
		0.270 × 0.5 × 4.76		0.643		
		0.270 × 0.5 × 3.76				0.508
H_2	0.025	0.025 × 0.5	0.0125			
		0.025 × 0.5 × 4.76		0.060		
		0.025 × 0.5 × 3.76				0.042
	1.000		0.1475	0.703	0.385	1.140

* Temperatures and pressures assumed to be the same as in the original composition

Air required = 0.703 cu. ft. per cu. ft. of gas; $CO_2 = 100 \times 0.385 / (0.385 + 1.14)$ = 25.2 per cent.

Properties of the Products of Combustion: Combustion, Dec., 1923, p. 451.

46. Flue-gas Analysis. — It has been shown that the products of combustion, commonly called flue gases, resulting from the complete oxidation of a fuel with theoretical air supply, consist chiefly of N_2 and CO_2, with small amounts of water vapor and SO_2. It was also shown that with a deficient air supply the flue gases may contain CO, H_2, and varying amounts of hydrocarbons. If air excess was used in the combustion process or air leaked into the setting, free O_2 would also be present in the gases. Evidently an analysis of the flue gases is an index to the amount of air furnished and, as will be shown later, is an important factor in determining the heat losses incident to combustion. The various appliances for sampling and analyzing the gaseous products of combustion are described in Chapter XVIII.

If a sample of flue gas shows the presence of only CO_2, N_2, H_2O and possibly SO_2, it is positive evidence that combustion is perfect. This result cannot be obtained in the commercial combustion of any fuel, because of the impossibility of completely oxidizing the combustible elements with the theoretical air supply.

If free O_2 is present with the CO_2, N_2, and SO_2, combustion is complete, but air has been supplied in excess of theoretical requirements. In practice, air excess is necessary, the minimum amount in an air-tight setting varying from 10 to 50 per cent or more, depending upon the nature of the fuel, the design of the furnace equipment, and the rate of combustion.

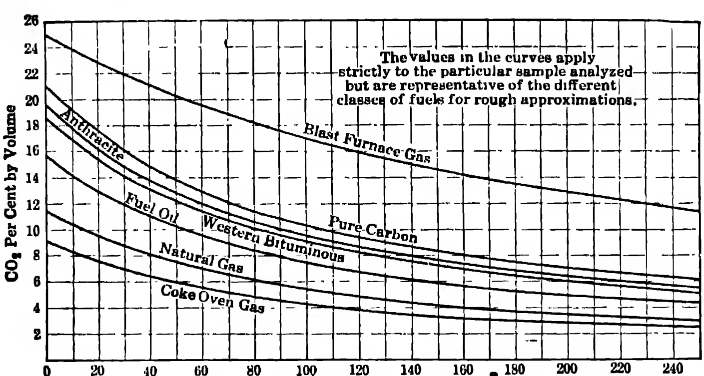

Fig. 14. Relation between Per Cent CO_2 by Volume in the Dry Flue Gases and Air Excess for the Complete Combustion of Typical Fuels.

For complete combustion with air excess, the percentage of CO_2 or O_2 is a true index of the excess, but it should be borne in mind that the maximum theoretical per cent of CO_2 is a function of the fuel itself and ranges from about 9 per cent for by-product coke-oven gas to over 25 per cent for dry blast-furnace gas. This is illustrated in Fig. 14.

If CO, H_2, or other combustibles are present in the flue gases, it is evident that combustion is incomplete, and if there is *no* free O_2, the air supply is deficient. However, the flue gases may contain CO, H_2, and other combustible gases, and considerable free O_2. This apparent anomaly is due to the fact that: (1) the combustible gases and the air supply have not been thoroughly mixed while in the zone of combustion; (2) the furnace temperature has been too low for proper ignition; or (3) there has not been sufficient time for combustion to be complete. Concentration, mixing, temperature, and time of control are interrelated, and each of these factors is essential for good combustion.

For each fuel there will be a definite percentage of the different constituents in the gaseous products incident to perfect combustion, but such percentages will vary not only for different classes of fuels, but even widely with different fuels of the same class. The actual volume of CO_2 resulting from the complete combustion of a specific fuel is constant, irrespective of the air excess, but the percentage by volume decreases as the excess increases. Furthermore, the actual volume of free oxygen and the percentage by volume increases with the amount of excess air; therefore, either the CO_2 or the free O_2 is a true index to the air excess for the complete combustion of the given fuel. If, however, combustion is incomplete, neither the percentage of CO_2 or that of O_2 is an accurate index to the air excess unless the character and the amount of the unburned combustible is known. In commercial boiler furnace practice, the unburned combustible in the flue gas is usually small in amount, so that either the O_2 or CO_2 is a fairly satisfactory index. The percentage of CO_2 is universally used as the index because of the ease with which it is obtained. The percentage of CO_2, by volume, in the dry gaseous products for complete combustion of several classes of fuels with varying air excess, is shown in Fig. 14.

Recent Developments in Flue Gas Analysis. Power, Sept. 18, 1923, p. 461.
Interpretation of Flue-gas Analysis: Combustion, Feb., 1924, p. 115.

47. Air Actually Supplied for Combustion. — In practice, the amount of air supplied is measured directly in situations where such measurements can be readily made, as in connection with mechanical draft, or where the entire air supply is forced to flow through a conduit, or the equivalent. Changes in the rate of flow may also be closely approximated by noting the pressure drop across the boiler. (See paragraph 343.) In most cases, however, physical measurements of flow are not feasible, and the amount of air supplied is calculated from the flue-gas analysis. The latter offers an accurate method for determining air excess, provided the sample of gas is truly representative of average conditions.

If CO_2', CO', O', N' = proportional part *by weight* of the carbon dioxide, carbon monoxide, free oxygen, and nitrogen in the dry flue gas, then the weight of carbon in the $CO_2 = 3/11\ CO_2'$ and that in the $CO = 3/7\ CO'$. The *weight G* of dry gas per lb. of carbon actually burned is

$$G = \frac{CO_2' + O' + CO' + N'}{3/11\ CO_2' + 3/7\ CO'} \tag{18}$$

If CO_2, CO, O, N = percentages by volume of these constituent gases, then $CO_2' = CO_2 \times 44/385$, $O' = O \times 32/385$ etc. (See equation 9.)

Substituting these values in equation (18) and reducing, we have[1]

$$G = \frac{11\ CO_2 + 8\ O + 7\ (CO + N)}{3\ (CO_2 + CO)} \tag{19}$$

Since $CO_2 + CO + O + N = 100$, neglecting traces of minor constituents, $CO = 100 - CO_2 - O - N$. Substituting this value of CO in the numerator of equation (19) and reducing, we have

$$G = \frac{4\ CO_2 + O + 700}{3\ (CO_2 + CO)} \tag{20}$$

Example 12. — Determine the weight of dry air supplied per lb. of coal as fired, analysis as in Example 9, if the flue gas resulting from the combustion is composed (per cent by volume) of

CO_2	12.8	O_2	5.4
CO	0.6	N_2	81.2

Solution. — Substitute the various percentages in equation (20) and solve, thus:

$$G = \frac{4 \times 12.8 + 5.4 + 700}{3\ (12.8 + 0.6)} = 18.82 \text{ lb. of dry gas per lb. of carbon}$$

actually burned.

Since the coal as fired contains 0.65 carbon, the dry gas per lb. of coal = $18.82 \times 0.65 = 12.23$ lb. If part of the coal falls through the grate, as is always the case in practice, the weight of carbon actually burned should be taken instead of the total carbon content.

The total weight of dry air actually supplied per lb. of coal burned is

$$12.23 - 0.65 + 8\ (0.044 - 0.072/8) = 11.86$$

[1] If SO_2 and free H are present in appreciable quantities, this expansion should have $16\ SO_2$ and $\frac{1}{2}$ H added to the numerator; if the hydrocarbons, C_2H_4 and CH_4 are also present $7\ C_2H_4$ and $4\ CH_4$ should be added to the numerator, and CH_4 and $2\ C_2H_4$ should be added to the parenthesis of the denominator. Except in special cases where the content of these gases is high, or where extreme accuracy is desired, it is common practice to disregard the presence of these constituents in calculating the air supply. The average flue-gas analysis is an approximation at the best, and refinement in calculation is without purpose.

It has been previously shown (Example 8) that the coal under consideration requires 8.87 lb. of air for theoretical combustion, hence,

$$\text{Air excess} = 100\ (11.86 - 8.87)/8.87 = 33.6 \text{ per cent.}$$

The 7 N in equation (19) represents the N supplied by the air less the negligible amount furnished by the coal itself. Since the nitrogen content of air is 77 per cent of the weight of the air, we have

$$A_2 = \frac{7\ N}{3\ (CO_2 + CO)} \div 0.77 = \frac{3.03\ N}{CO_2 + CO} \qquad (21)$$

in which

A_2 = the weight of dry air supplied per pound of *carbon burned*.

N, CO_2, CO = percentages by volume of nitrogen, carbon dioxide and carbon monoxide in the flue gas.

For the example cited above:

$$A_2 = 3.03 \times 81.2 \div (12.8 + 0.6) = 18.36 \text{ lb.}$$

For the coal under consideration:

$$\text{Dry air per lb.} = 0.65 \times 18.36 = 11.93$$

This checks approximately with results calculated from equation (20).

The term 7 N in equation (19) neglects the nitrogen content of the fuel itself, and for this reason the formula is not applicable to fuels high in nitrogen, as for example, blast-furnace gas.

The percentage of CO_2 by volume, in the dry gaseous products, for complete combustion of several classes of fuels with varying air excess is shown in Fig. 14.

For a given furnace and a given fuel, there is a definite air excess which gives the maximum overall commercial efficiency, but this can be determined only by actual service test.

Effect of Air Excess on Flue Temperatures and on Efficiency: Power, Apr. 22, 1923, p. 634.

48. Temperature of Combustion. — The actual temperature of the furnace, fuel bed, or any other part of the furnace equipment is most satisfactorily determined by means of a suitable pyrometer. Great care, however, must be used in making such measurements, as shown in Bul. 145, U. S. Bureau of Mines, 1918, by Kreisinger and Barkley. The maximum theoretical temperature resulting from the combustion of any fuel may be calculated from the simple relationship

$$\text{Heat absorbed} = \text{heat given up}$$

Assuming that all the heat generated is absorbed by the products of combustion, this relationship may be expressed

$$wc\,(t_1 - t) = H \tag{22}$$

in which

w = weight of the products of combustion, lb. per lb. of fuel
c = mean specific heat of the products of combustion
t_1 = final temperature of the products of combustion, deg. fahr.
t = initial temperature of the fuel and air supply, deg. fahr.
H = heat actually liberated by the combustion of the fuel, B.t.u. per lb.

The final temperature, t_1, as calculated from equation (22) is purely hypothetical and can never be realized in practice, because no apparatus has been constructed which permits all of the heat liberated to be absorbed by the products of combustion. In any kind of an enclosed furnace — as in a reverberatory, or a blast furnace, or beehive oven — temperatures calculated by means of equation (22) are much nearer the actual value than those found in boiler furnaces, chiefly on account of the heat radiated from the fuel bed or furnace walls to the cooler surrounding surfaces. By including a suitable factor for radiation, equation (22) may be used for approximating the actual temperature of combustion; but attention should be called to the fact that this factor (as well as w, c, t, and H in equation (22)), is the product of many variables and requires careful analysis for even approximate results. If r = heat radiated to and absorbed by the surrounding cooler surfaces, B.t.u. per lb. of fuel, equation (22) may be expressed

$$wc\,(t_1 - t) + r = H \tag{23}$$

The weight of the gaseous products of combustion may be calculated from the flue-gas analysis, as shown in paragraph 46, or it may be predetermined for any assumed air excess and completeness of combustion, as shown in paragraph 45. The mean specific heat may be closely approximated, as shown in the latter part of this paragraph. The heat actually liberated by the combustion of the fuel may be obtained with a fair degree of accuracy by subtracting the heat loss due to incomplete combustion from the total heat value of the fuel. The amount of heat transmitted by radiation may be roughly approximated from equation (42).

Example 13. — Required the theoretical temperature of combustion of carbon in air, if 50 per cent air excess is necessary for complete combustion and there is no radiation or other loss; initial temperature of air and fuel, 60 deg. fahr. Required also the amount of heat transmitted by

radiation if the actual temperature is 500 degrees less than the theoretical maximum.

Solution. — One lb. of carbon requires 11.58 lb. of air for complete combustion without air excess; therefore, the weight of the gaseous products for 50 per cent air excess = $1.5 \times 11.58 + 1 = 18.37$ lb. Substituting $w = 18.37$, $c = 0.265$ (assumed), $t = 60$ in equation (22) and reducing

$$18.37 \times 0.265\,(t_1 - 60) = 14{,}600$$
$$t_1 = 3100 \text{ deg. fahr. (approx.)}$$

Since the actual temperature = 3100 − 500 = 2600, the amount of heat transmitted by radiation may be calculated by substituting this value for r in equation (23) thus:

$$18.37 \times 0.265\,(2600 - 60) + r = 14{,}600$$
$$r = 2235 \text{ B.t.u. per lb. of carbon, or}$$

15.3 per cent of the calorific value of the fuel.

In modern stoker-fired furnaces where a large portion of the boiler heating surface is exposed to radiation, as much as 50 per cent of the total heat absorbed by the boiler is transmitted through radiation. In enclosed furnaces of the "Dutch Oven" type, this quantity is frequently less than 10 per cent.

In practice, the maximum furnace temperature is limited by the softening point of the refractories and the formation of objectionable clinker. In order to maintain a safe fuel-bed temperature, and at the same time reduce the amount of air excess to a minimum, a large portion of the boiler heating surface is exposed to direct radiation from the fuel bed.

For an excellent article on the relation between furnace temperatures, excess air, and the rate of combustion, see "Combustion of Coal," by R. B. MacMullin, *Combustion*, May 1922, p. 224.

Data relative to the specific heats of gases are rather discordant. The following equations are considered to be as nearly accurate as any. (See Principles of Thermodynamics, G. A. Goodenough, 3rd. ed. p. 275.)

For N_2 and CO, $C = 0.246 + 0.000{,}009\,t$ (24)

O_2 $C = 0.215 + 0.000{,}008\,t$ (25)

H_2 $C \doteq 3.44 + 0.000{,}135\,t$ (26)

Air $C = 0.238 + 0.000{,}0086t$ (27)

CO_2 $C = 0.2 + 0.000{,}0365t - 0.000{,}000{,}004\,t^2$ (28)

H_2O $C = 0.433 + 0.000{,}0432t + 0.000{,}000{,}0023\,t^2$ (29)

C = mean specific heat at constant pressure between zero and t deg. fahr.

Between 2000 deg. fahr. and 2800 deg. fahr., the results are uncertain, and dependence can be placed only in the first two significant figures in

the decimal. Beyond 2800 deg. fahr., the results are purely conjectural, since experiments have not been made at these high temperatures.

If the mean specific heats, $c_1\ c_2\ .\ .\ .\ c_n$, and weights $w_1\ w_2\ .\ .\ .\ w_n$, of the constituent gases of a compound are known, the mean specific heat of the compound may be determined as follows:

$$c = \frac{w_1c_1 + w_2c_2\ .\ .\ .\ .\ + w_nc_n}{w_1 + w_2\ .\ .\ .\ .\ + w_n} \tag{30}$$

The application of equations (24–29) at high temperatures to equations (22–23) necessitates laborious calculations; and since the results are only approximate at the best, extreme refinement in calculation is without purpose. The curves in Fig. 15 are plotted from these equations and afford a means of approximating the specific heat without the labor of solving the equations.

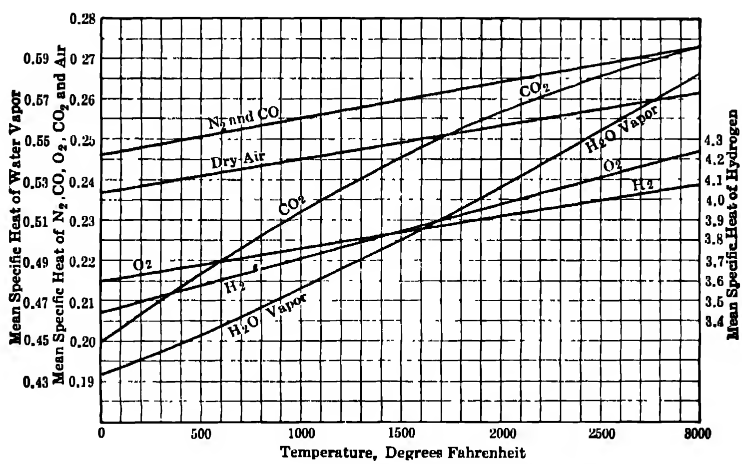

Fig. 15. Mean Specific Heat of Gases.

49. Losses in Burning Fuels. — A boiler, in order to entirely utilize the heat of combustion of the fuel, must be free from radiation and leakage losses, the fuel must be completely burned, and the products of combustion must be discharged to waste at a temperature somewhat below that of the initial fuel and air supply. While it is possible to design a boiler and furnace to effect this result within 1 per cent or so of perfection, such an installation would not be a commercial success with fuels at the present price, because of high first cost and cost of operation and maintenance. A boiler which absorbs 85 per cent of the heat value is exceptional, and an average figure for very good practice is not far from 80 per cent. The

various losses, including the heat utilized, constitute the boiler and furnace "heat balance." The losses usually considered are:

Developed heat discharged through stack

1. Dry chimney gases
2. Moisture in fuel
3. Moisture from combustion of hydrogen
4. Moisture in the air

Fuel losses

a. Loss due to carbon monoxide
b. Loss of fuel through grate
c. Unburned fuel (other than carbon monoxide)

Radiation and unaccounted for.

Some of these losses are preventable; others are inherent and cannot be avoided.

The heat losses incident to the combustion of fuels in boiler units are the products of many variables; and while it is possible to establish empirical rules which give satisfactory results within a limited range, the rational calculations are comparatively simple and therefore no attempt will be made to include empirical factors. Considerable time, however, may be saved by substituting constants for variables and reducing the equation to its simplest form, provided the variation in the assumed values for the constants is not sufficiently great to seriously affect the results.

50. Sensible Heat Lost in the Dry Chimney Gases. — This loss depends upon the nature of the fuel, type and proportions of boiler, furnace and setting, and upon the rate of driving. It is usually the greatest of all the losses. The heat carried away may be expressed:

$$h_1 = w\ (t_c - t)\ c, \tag{31}$$

in which

h_1 = B.t.u. lost per lb. of fuel
w = weight of dry chimney gases per lb. of fuel
t_c = temperature of the escaping gases, deg. fahr.
t = temperature of the air entering the furnace
c = mean specific heat of the dry gases. (This may be taken as 0.24 for most purposes.)

A glance at equation (31) will show that this loss may be reduced (1) by decreasing the weight of the products of combustion and (2) by reducing the temperature difference between the air entering the furnace and the flue gas leaving the boiler.

To reduce the weight of the products of combustion, the air excess must be lowered. For a given furnace and boiler equipment, and fuel, there is a definite air excess which gives the maximum overall efficiency, but this can be determined only by actual service test. With gaseous, liquid, and powdered fuels, this excess may be kept within 10 to 20 per cent over theoretical air requirements, but with bulk fuels, the minimum excess is seldom less than 35 per cent, except in connection with scientifically operated large plants. In the average plant, the excess ranges from 50 to 200 per cent or more.

The temperature difference between the air supply and flue gas may be reduced by preheating the air, or by lowering the flue gas temperature, or both. Unfortunately, this temperature difference for a given equipment and fuel is a function of the air excess and rate of driving, and therefore cannot be reduced *per se*. This is on the assumption, of course, that neither economizer nor air preheater element is installed.

It is possible to abstract as much heat from the flue-gases by the air preheater as with the economizer. By using air in this manner, the maximum opportunity for raising the turbine room efficiency is offered, and in the boiler room the same heat from the flue-gases can be recovered but with additional intermediate improvement in combustion efficiency. Combustion reactions are greatly accelerated by increasing the temperature, and with preheated air it is therefore possible to more nearly obtain complete combustion with lower percentage of air both on the grate and in the combustion chamber. Preheating air not only increases efficiency but makes possible higher rate of combustion with low ashpit losses. See paragraph 264.

Considering the temperature of the air supply as atmospheric, which is the case in the great majority of installations, the temperature difference can be reduced only by lowering that of the flue gas. This may be readily accomplished through the use of economizers (see paragraph 261), but for the boiler proper, air excess above the minimum for best efficiency usually results in increased flue-gas temperatures, though, of course, it is possible to carry this dilution to a point where the flue-gas temperature will be lowered. The theory involved in this apparent anomaly includes such factors as heat-transfer rates, difference in temperature between the gases and the absorbing surface, the percentage of the total heat absorbed through radiation and convection, and so on; but the fact remains that for a given equipment, fuel, and load, air excess above a certain minimum results in *increased* flue-gas temperatures. A high percentage of CO_2 is frequently accompanied

[1] *Preheat and Excess Air on Combustion of Fuel:* Power, June 14, 1921, p. 960; Aug. 30, 1921, p. 339; Nov. 29, 1921, p. 844; Apr. 22, 1924, p. 634.

by a high CO content, sometimes leading to secondary or delayed combustion between the boiler tubes, and resulting in increased temperature of the flue gases. The temperature of the flue gas cannot be made lower than that of the heat-absorbing surface with which it was last in contact, and, as a matter of fact, ranges anywhere from 15 to 400 degrees above that minimum. See Figs. 55 and 65.

The heat discharged by the dry chimney gases ranges from 3.8 per cent of the total heat generated in the best recorded performance of a powdered-coal plant with economizer, to 30 per cent or more in poorly operated bulk-fuel plants. See typical heat balances, Tables 18 and 19. The curves in Fig. 16 give the dry chimney-gas loss for different fuels

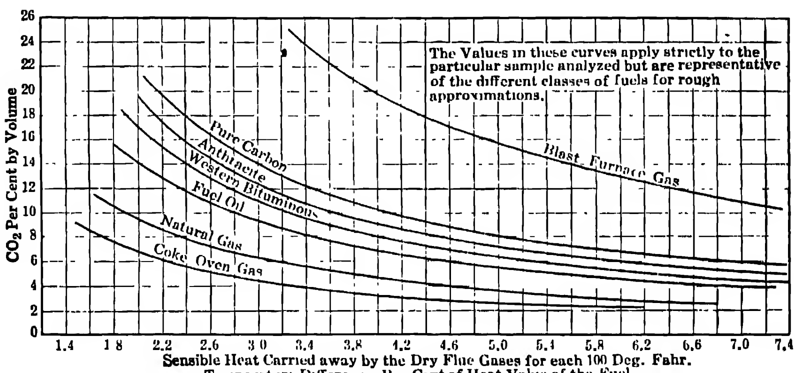

FIG. 16. Relation Between the Sensible Heat Loss in the Dry Flue Gases and the Per Cent of CO_2 for Complete Combustion of Typical Fuels.

with varying degrees of air excess, as indicated by the CO_2 content for each 100 degrees difference in temperature between that of the air supply and the flue gas.

Example 14. — Calculate the sensible heat loss in the dry chimney gases if pure carbon is completely burned with 50 per cent air excess; initial temperature 60 deg. fahr; flue-gas temperature 460 deg. fahr.

Solution. — The weight of dry chimney gases for the complete combustion of pure carbon with 50 per cent air excess is $1.5 \times 11.58 + 1 = 18.37$ lb. per lb. of carbon. Substituting $w = 18.37$, $t = 60$, $t_c = 460$ in equation (31) and reducing we have

$$h_1 = 18.37\ (460 - 60)\ 0.24 = 1763 \text{ B.t.u. per lb.}$$

$$1763 \div 14{,}600 = 0.12 = 12 \text{ per cent of the heat value of the carbon.}$$

51. Heat Loss Due to Evaporating the Moisture in the Fuel. — Moisture in fuel reduces the efficiency of the steam-generating apparatus by discharging heat up the stack in the form of highly superheated steam.

Except with green fuels very high in moisture content, such as bagasse, tanbark, wood and the like, the heat loss is of small consequence. On the other hand, many coals burn to better advantage when properly tempered (moistened) than when burned dry. According to tests conducted by T. A. Marsh[1] with Western coals, (1) properly tempered coal has less resistance to air flow than either very wet or very dry coal, (2) very wet coal has less resistance to air flow than very dry coal, (3) properly tempered coal burns to a cleaner ash by decreasing the fuel-bed resistance, and (4) properly tempered coal causes less siftings than dry coal and is productive of fewer holes in the fire bed. Exhaust steam for tempering purposes gives very satisfactory results and is extensively used. See also paragraph 103.

The loss due to evaporating the moisture may be expressed

$$h_2 = w\,[H - c_1\,(t - 32) + c'\,(t_c - t')] \tag{32}$$

in which

h_2 = B.t.u. lost per lb. of fuel,
w = weight of free moisture per lb. of fuel,
H = total heat of 1 lb. of saturated steam above 32 deg. fahr., corresponding to the temperature at which evaporation takes place.
c_1 = mean specific heat of water, 32 to t deg. fahr.
t = temperature of the fuel, deg. fahr.,
c' = mean specific heat of the water vapor, t_c to t deg. fahr.,
t_c = temperature of the chimney gas,
t' = temperature at which evaporation takes place, deg. fahr.

The temperature at which evaporation begins is low because of the low partial pressure of the vapor in the gaseous products of combustion and may range from 70 to 120 deg., depending upon the composition of the gases and the amount of moisture evaporated. Fortunately, the term $H - c't'$ is practically constant for a wide range of t' and consequently a knowledge of the actual value for each set of conditions is not necessary for the purpose at hand.

Assuming $c' = 0.455$, $c_1 = 1$ and taking H from the steam tables for all values ranging from $t' = 70$ to $t' = 120$ deg., we find that $H - 0.455\,t' = 1058.7$. Substituting this value in equation (32) and reducing

$$h_2 = w\,(1090.7 - t + 0.455\,t_c) \tag{33}$$

The loss due to superheating the free moisture under average boiler operating conditions is approximately 12.5 B.t.u. for each per cent of moisture. Thus with wood waste containing 50 per cent moisture, the

[1] Power Plant Engineering, Feb. 15, 1923, p. 215: Combustion, Jan. 1924, p. 37.

loss would be about 15 per cent of the heat value of the fuel as fired. With ordinary good coal, the loss seldom exceeds 0.5 per cent of the heat value of the fuel as fired.

What Happens to Moisture When it Enters the Furnace: Power, Oct. 30, 1923, p. 700; Dec. 18, 1923, p. 1001.

52. Loss Due to the Combustion of Hydrogen in the Fuel. — The hydrogen (other than that in the free moisture) in any fuel burns to water, and in so doing liberates a certain amount of heat. All of this heat is not available for producing steam in the boiler, since the water formed by combustion is discharged with the flue gases as superheated steam at chimney temperature. This loss is equal to

$$h_3 = 9H\ (1090.7 - t + 0.455\ t_c) \tag{34}$$

in which

h_3 = B.t.u. lost per lb. of fuel
H = weight of hydrogen per lb. of fuel.

All other notations as in equation (32) and (33).

With anthracite coal this loss is approximately 2.5 per cent of the total heat value of the combustible, and with bituminous coal it runs as high as 4.5 per cent. With fuel oil the loss is approximately 6.5 per cent, and with coke-oven gas about 14 per cent of the heat value of the fuel. With economizers, these losses may be reduced considerably.

53. Heat Loss Due to Superheating the Moisture in the Air. — The loss due to this cause is a minor one, though on hot humid days it may be appreciable. Except in very carefully conducted boiler tests, it may be disregarded, since its value is usually less than the errors of observation of many of the influencing factors. This loss may be expressed:

$$h_4 = wc\ (t_c - t) \tag{35}$$

in which

h_4 = B.t.u. lost per lb. of fuel,
w = weight of moisture introduced with the air per lb. of fuel,
c = mean specific heat of water vapor, t to t_c deg. fahr.,
t = temperature of air entering the furnace, deg. fahr.,
t_c = temperature of chimney gases, deg. fahr.

$$w = zdvA \tag{36}$$

in which

z = relative humidity (see paragraph 409),
d = weight of 1 cu. ft. of water vapor at t deg. fahr. (this may be taken directly from steam tables).

v = volume of 1 lb. of dry air plus its moisture content at t deg. fahr., cu. ft. (for the purpose at hand this may be taken as the volume of 1 lb. of dry air).
A = weight of dry air supplied per lb. of fuel.

This loss seldom exceeds 0.5 per cent of the heat value of the combustible portion of the fuel and is ordinarily less than 0.3 per cent.

Example 15. — Calculate the heat lost in superheating the moisture in air under the following conditions: Temperature of air entering furnace, 100 deg. fahr.; temperature of flue gases, 550 deg. fahr.; relative humidity of air entering furnace, 40 per cent; weight of air supplied per lb. of fuel, 18 lb.; heat value of the fuel fired, 12,500 B.t.u. per lb.; standard atmospheric pressure.

Solution. — From steam tables for $t = 100$, we find $d = 0.00285$ lb. and by means of equation (8) we calculate $v = 14.11$ cu. ft., $A = 18$ lb., and $z = 0.40$ as stated. Substituting these values in equation (36) and reducing, we have $w = 0.40 \times 0.00285 \times 14.11 \times 18 = 0.29$.

c may be taken as 0.46. Substituting $c = 0.46$, $t_c = 550$ and $t = 100$ in equation (35) and reducing we have

$$\begin{aligned} h_4 &= 0.29 \times 0.46\ (550 - 100) \\ &= 60 \text{ B.t.u. per lb. of fuel.} \end{aligned}$$

$60 \div 12{,}500 = 0.005 = 0.5$ of 1 per cent of the heat value of the fuel.

54. Loss Due to Carbon Monoxide. — In the absence of sufficient oxygen for their complete combustion, the complex hydrocarbons, constituting the volatile matter leaving the fuel bed, are quickly decomposed by the high furnace temperature into soot, H_2 and CO. The formation of the latter is due to the presence of CO_2 and the small supply of oxygen. The persistence of CO in the furnace gases is due to the constant reaction between soot, H_2 and CO_2, and not to the difficulty of burning it. CO comprises approximately 80 per cent of the combustible gases; the other 20 per cent consists mainly of hydrogen and a trace of methane (CH_4). Unless sufficient oxygen is present to completely oxidize the products of distillation within the combustion zone, or the mixture of air and gases is not thorough, or the temperature is below the ignition point of the gases, some of the carbon may escape as CO. The presence of even a small amount of CO in the flue gas is indicative of an appreciable heat loss, as will be seen from Table 13. CO is invisible, and its presence in the flue gases therefore cannot be detected except by analysis.

If C is the proportional part of the carbon in the fuel which is *actually burned*, and $CO_2' + CO'$ = percentage by *weight* of the CO_2 and CO in the dry flue gas, then the weight of carbon in the $CO_2 = 3/11\ CO_2'$, and

that in the CO = 3/7 CO′, and the weight w_{co} of CO per lb. of fuel as fired is

$$W_{co} = C\frac{CO'}{3/11\ CO_2' + 3/7\ CO'} \tag{37}$$

If CO_2 and CO = percentage by *volume* of these constituents, then $CO_2' = CO_2 \times 44/385$ and $CO' = CO \times 28/385$. (See equation 7.) Substituting these values in equation (37) and reducing, we have

$$W_{co} = C\frac{7\ CO}{3\ (CO_2 + CO)} \tag{38}$$

But 3/7 of the weight of the monoxide is carbon; therefore, the weight of *carbon*, w_c, in the CO content of the flue gas, per lb. of fuel as fired is 3/7 w_{co} or

$$w_c = C\frac{CO}{CO_2 + CO} \tag{39}$$

Since the heat of combustion of C to CO is 4,440 B.t.u. against 14,600 B.t.u. for complete combustion of C to CO_2, the heat loss may be expressed

$$h_5 = w_c\ (14{,}600 - 4{,}440) = C\frac{10{,}160\ CO}{CO_2 + CO} \tag{40}$$

in which

h_5 = heat loss due to the escape of CO, B.t.u. per lb. of fuel as fired.

Other notations as previously defined.

This loss may be reduced to a negligible quantity in a properly designed and carefully operated furnace. In fact the loss from this cause is often exaggerated and seldom exceeds 1 per cent of the total heat value of the fuel except during the few moments following the replenishing of a burned-down fire with fresh fuel or when the supply of air is checked to meet a sudden reduction in load. In improperly designed furnaces in which the volatile gases and air supply are not thoroughly mixed before leaving the furnace, or where the temperature of the products of combustion is reduced below the ignition point of the gases before oxidation is complete, a considerable amount of CO may escape unburned, and in such a case the loss may prove to be a serious one.

High efficiencies necessitate minimum air excess; hence the presence of a small amount of CO may be expected in the flue gas for high percentages of CO_2. In a number of recent tests of modern central station boilers operating 150 to 350 per cent of standard rating, the loss due to the escape of CO in the flue gas ranged from 0.2 to 1.95 per cent of the heat value of the fuel (western bituminous) with a general average, extended

TABLE 13

LOSS DUE TO INCOMPLETE COMBUSTION OF PURE CARBON TO CARBON MONOXIDE
Per Cent of the Calorific Value of Carbon

Per Cent by Volume of CO in the Dry Flue Gas	Per Cent by Volume of CO_2 in the Dry Flue Gas							
	4	6	8	10	12	14	16	18
0.1	1.70	1.15	0.86	0.59	0.53	0.49	0.43	0.39
0.2	3.33	2.26	1.70	1.37	1.15	0.99	0.86	0.77
0.3	4.89	3.33	2.53	2.07	1.70	1.47	1.29	1.15
0.4	6.37	4.35	3.33	2.69	2.26	1.94	1.70	1.37
0.5	7.79	5.38	4.12	3.33	2.80	2.41	2.12	1.89
0.6	9.41	6.36	4.88	3.96	3.33	2.87	2.53	2.26
0.7	......	7.30	5.14	4.58	3.86	3.33	2.93	2.64
0.8	......	8.24	6.36	5.18	4.37	3.78	3.33	2.98
0.9	....	9.14	7.08	5.28	4.88	4.23	3.73	3.33
1.0	....	......	7.78	6.36	5.38	4.66	4.22	3.68
1.2	....	..	9.12	7.50	6.36	5.62	4.88	4.37
1.4	.	.....		8.59	7.31	6.36	5.63	5.05
1.6		......	..	9.65	8.22	7.18	6.36	5.70
1.8	....	....	.	..	9.14	7.98	7.08	6.37
2.0	.....	.....		.. .		8.75	7.78	7.00

Heat value of carbon assumed to be 14,600 B.t.u. per lb.
The entire carbon content of the fuel assumed to be burned to CO and CO_2.

over several days, of 0.4 per cent. In these tests the per cent of CO_2 in the flue gas ranged from 11.95 to 15.45. The CO content appears to increase with the increase in CO_2 and furnace temperature, as shown in

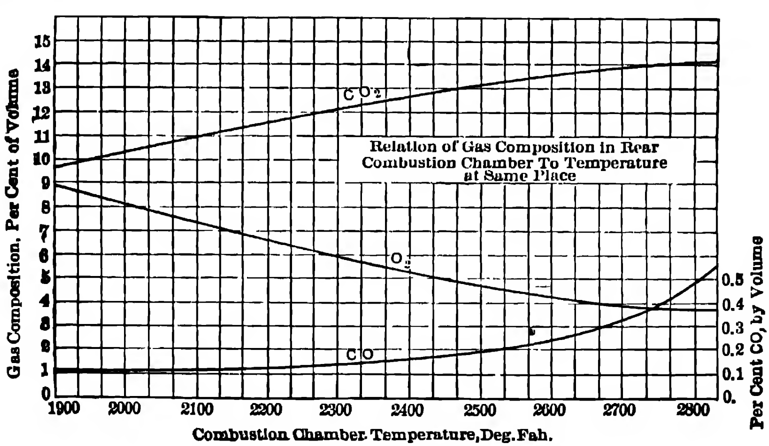

FIG. 17. Relation of Gas Composition to Temperature.

Fig. 17, the curves of which are based on tests of a 250-hp. Heine boiler, hand-fired. Almost complete absence of CO is to be expected with moderate air excess in any well-designed furnace; but it is possible for a

high percentage of CO and a great excess of air supply to exist at the same time, though this combination is not likely to occur in a properly designed and correctly operated furnace except at very low rates of combustion. See Table 14.

TABLE 14

RELATION OF CO AND COMBUSTION-CHAMBER TEMPERATURES

(U. S. Geological Survey)

	Per Cent of Black Smoke						
	0	0 to 10	10 to 20	20 to 30	30 to 40	40 to 50	50 to 60
Number of tests ...	37	18	56	51	36	17	4
Average per cent of smoke	0	7.1	15.5	24.7	34.7	43.1	52.9
Average per cent of CO in flue gases	0.05	0.11	0.11	0.14	0.21	0.33	0.35
Average per cent unaccounted for in heat balance	9.14	10.60	9.46	10.93	11.41	13.41	13.34
Number of tests*	26	16	48	45	32	17	4
Average combustion-chamber temperature (deg. fahr.)	2180	2215	2357	2415	2450	2465	2617

* Temperatures in combustion chamber were not determined on all tests.

Example 16. — Calculate the heat loss due to the escape of CO in the flue gas for the following conditions: Per cent CO and CO_2 by volume in the flue gas, 0.6 and 12.8, respectively; analysis of coal as fired — C 65, ash 0.13, B.t.u. per lb. 11,850; combustible in dry refuse, 20 per cent.

Solution. — Carbon actually burned = $0.65 - 20 \times 13/(100 - 20) =$ 0.6175 lb. per lb. of coal as fired. Substituting this value for C in equation 40, noting that CO = 0.6 and CO_2 = 12.8, and solving

$$H_5 = 0.6175 \times 10{,}160 \times 0.6/(12.8 + 0.6) = 281 \text{ B.t.u. per lb.}$$

$281 \div 11{,}850 = 0.0237$ or 2.37 per cent of the heat value of the coal as fired.

55. Loss of Fuel Through Grate. — The refuse from a fuel is that portion which falls into the pit in the form of ashes, unburned or partially burned fuel and cinders.

In steam boiler practice the unconsumed carbon in the ashpit ranges from 10 to 50 per cent of the total weight of dry refuse, depending upon the size and quality of coal, type of grate, and rate of driving. The loss resulting from this waste of fuel ranges from 1.5 to 10 per cent or more, of the heat value of the fuel. It is impossible to assign a minimum value because of the various influencing factors, but numerous tests of recent installations, equipped with mechanical stokers, indicate that actual loss

ranges from 1.5 to 5 per cent of the heat value of the fuel at normal driving rates. Coal which necessitates frequent slicing is apt to give greater losses from this cause than a free-burning coal.

Extensive tests conducted by the American Gas & Electric Co. (Reginald Trautschold, *Power*, Feb. 22, 1916, p. 256) show that the actual yearly loss due to combustible in the refuse is not directly proportional to the combustible content, but increases as shown by the "actual loss" curve in Fig. 18. Thus, the reduction of the combustible content from 10 per cent to 5 per cent effects a yearly saving in the ratio of 12.98 to 5.83 instead of 10 to 5. The percentage of combustible in the refuse also appears to increase with the increase in initial ash content.

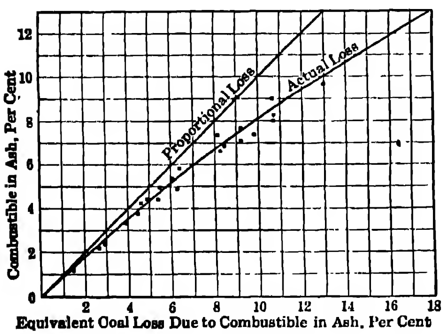

Fig. 18. Coal Loss Due to Combustible in Ash.

In some types of natural-draft traveling grates in which a large percentage of the fine fuel falls through the front end of the grate, a special hopper is ordinarily installed in the ashpit which reclaims most of it. (See Fig. 167.)

If h_c = calorific value of combustible in the dry refuse,
y = percentage of combustible in the dry refuse,
a = percentage of ash in the coal as fired,
h_6 = heat loss in the refuse, B.t.u. per lb. of coal as fired.

Then, assuming that all the ash in the fuel as fired appears in the ashpit, the weight of combustible in the ashpit may be expressed as $ya \div 100(100 - y)$ lb. per lb. of fuel as fired, and the heat loss as

$$h_6 = \frac{h_c}{100}\left(\frac{ya}{100 - y}\right) \tag{41}$$

For the average boiler test, the calorific value of the combustible in the dry refuse may be taken as that of pure carbon, but for accurate results calorimetric determinations are necessary.

Equation (41) should not be used if an appreciable amount of the ash is deposited throughout the setting or discharged through the stack. In this case the refuse in the ashpit should be weighed and analyzed for combustible.

Loss Due to Carbon in Furnace Refuse: Power, Sept. 23, 1919, p. 500.
Ashpit Losses: Combustion, Aug., 1923, p. 175.

56. Loss Due to Visible Smoke. — Soot is formed by the incomplete combustion of the hydrocarbon constituents of a fuel. All hydrocarbons are unstable at furnace temperatures, and unless air to insure complete combustion is mixed with them at the time they are distilled, they are quickly decomposed, the ultimate product consisting mostly of soot, H_2 and CO. Soot is formed at the surface of the fuel bed by heating the hydrocarbons in absence of air; it is not formed by the hydrocarbons striking the comparatively cool heating surface of the boiler. As a matter of fact, only a small trace of hydrocarbon gases reaches the boiler heating surface, provided there is a supply of air above the fire; hydrocarbons that do so are prevented from decomposition by the reduction in temperature due to contact. Once formed, it is difficult to burn it in the atmosphere of the furnace, because the oxygen is greatly rarefied, the gases containing only a few per cent of free oxygen.

Experience with burning soft coal shows that, if soot is once formed, a large percentage remains floating in the gases after all the other gaseous combustibles have been completely burned. Part of the soot is deposited on the tubes and throughout the boiler setting, while the rest is discharged through the stack with the gaseous products of combustion. A smoky chimney does not necessarily indicate an inefficient furnace, since the fuel loss due to visible smoke seldom exceeds 1 per cent. See Table 15. As a matter of fact, a smoky chimney may be much more economical than one that is smokeless. Thus, a furnace operating with very small air excess may cause considerable visible smoke and still give a higher evaporation than one made smokeless by a very large air excess. There will be some loss due to CO, unburned hydrocarbons, and soot in the former case, but in the latter this may be offset by the excessive loss caused by the heat carried away in the chimney gases. In general, however, smoky chimneys indicate serious losses, not because of the soot, but on account of the unburned, invisible combustible gases. (See Table 17.) The loss under this paragraph heading refers strictly to the visible combustible discharged up the stack and not that deposited on the tubes and in various parts of the setting. With natural draft the latter seldom exceeds a fraction of 1 per cent of the heat value of the fuel. In case of very high rate of combustion under forced draft, the loss due to combustible in the cinders may range as high as 10 per cent or more. A well-designed furnace, properly operated, will burn many coals without smoke up to a certain rate of combustion. Further increase in the amount burned will result in smoke and lower efficiency due to deficient furnace capacity. Small sizes of coal ordinarily burn with less smoke than larger sizes, but develop lower capacities. In the average hand-fired furnace, washed coal burns with lower efficiency and makes more smoke than raw

coal. Most coals that do not clinker excessively can be burned with a smaller percentage of black smoke than those which clinker badly. For means of determining smoke density, see paragraph 354.

TABLE 15

QUANTITY AND HEAT VALUE OF SOLIDS IN VISIBLE SMOKE

(Bituminous Coal)

From the Report of the Chicago Association of Commerce Committee of Investigation on Smoke Abatement. (1912)

Test Number	Smoke Density Per Cent	Solids in Visible Smoke	
		Per Cent by Weight of Fuel Fired	Per Cent of the Heat Value of the Fuel Fired
Fires with High Smoke Density			
3	21 97	0 83	0 28
17	20 00	0.75	0 36
10	20 00	1 10	0.95
30	15.80	0 65	0 49
29	14.50	0 82	0 49
Average	18 45	0 63	0 51
Fires with Low Smoke Density			
56	0	0 51	0 21
57	0	0.30	0 08
80	0	4.07	0 74
81	0	1 81	0 48
85	0	0.47	0.11
Average	0	0.47*	0.32

* Average of 10 plant tests not including Test No. 80.

TABLE 16

CHEMICAL COMPOSITION OF THE SOLID CONSTITUENTS OF SMOKE

(Chicago Association of Commerce)

Kind of Fuel	Per Cent of Total Solids				
	Hydrocarbons (Tar)	Combustible Solids (Carbon)	Mineral Matter (Ash)	Sulphur	Total
High-pressure Plants					
Pocahontas...	3 08	41.45	52.39	3 08	100
Bituminous..	4 19	32 80	59.93	3 08	100
Low-pressure Plants					
Anthracite..........	0.73	31.88	67 39	0 0	100
Pocahontas.........	11.43	54.90	33.47	0.20	100
Bituminous..........	31.43	44.06	22.12	2 39	100

TABLE 17

ANALYSIS OF CHIMNEY GASES

Boiler	Smoky						Clear					
	CO_2	O_2	CO	CH_4	H_2	N_2	CO_2	O_2	CO	CH_4	H_2	N_2
No. 1, hand-fired..	11 00	6 90	0 90			81 20						
	10 65	6 45	2 15			80 75						
No. 1, with smoke-prevention device							7 00	13 50	0			79 50
							9 00	9 75	0			81 25
No. 2, hand-fired	10 25	8 60	50	0	0	80 65						
No. 3, hand-fired	13 25	3 50	05	0 25	0	82 95						
No. 4, fire under caustic pot, hand-fired	10 95	1 30	3 00	70	3 23	80 82						
No. 5, split bridge, hand-fired	8 75	7 00	3 25	.40	1 00	79 60						
No. 6, with smoke-prevention device	..				....	...	7 25	12 00	0	0	0	80 75
No 7, with smoke-prevention device							7 15	12 15	0	0	0	80 70
No. 8, with smoke-prevention device							8 15	11 10	0	0	0	80 75

57. Radiation and Unaccounted For. — These losses are usually determined by difference. That is, the difference between the heat represented in the steam and the sum of the losses just mentioned is charged to "unconsumed hydrogen and hydrocarbons, to radiation, and unaccounted for." Unless accurate observations have been made in determining the various factors entering into the heat balance, the "radiation and unaccounted for" loss may represent a large percentage of the total heating value of the coal. Careful tests on *well-designed* boiler furnaces show that the radiation loss seldom exceeds 2 per cent. In case of very poorly installed settings or when the rate of driving is very low, the radiation loss may be considerably more than this. An examination of the data from carefully conducted tests of modern boiler furnaces will show that the "radiation and unaccounted for" items range from 2 to 6 per cent with an average of about 4 per cent. Soot deposited on the boiler tubes and throughout the setting, and cinders blown out of the stack under high draft pressures may greatly increase the unaccounted for loss, unless means are available for determining these factors.

58. Heat Balance. — Any chart giving the distribution of the various heat items constitutes a heat balance. The greater the number of subdivisions the more readily is it possible to locate the source of loss. In everyday furnace practice a determination of the heat balance is seldom

attempted because of the expense and difficulty of obtaining the various factors entering into the calculations; and even if the expense is of secondary consideration, the results are apt to be more or less approximate. This is particularly true in situations where the load is constantly fluctuating. In general practice the operating engineer is chiefly interested in the heat absorbed by the boiler (as shown by the evaporation per pound of fuel as fired) and the relative sensible heat loss up the stack (as indicated by the percentage of CO_2 in the flue gas and the temperature of the gases in the uptake). The various factors entering into the commercial boiler heat balance, as recommended by the 1915 Code of the American Society of Mechanical Engineers,[1] are itemized in Table 18. According to this code, the heat distribution is expressed in terms of " dry coal " or " combustible." When the performance of different installations is to be compared, this offers a most satisfactory basis; but the operating engineer, in tracing out the source of heat loss with a view of bettering operation, is chiefly concerned with " coal as fired " and for this reason the heat balance is commonly expressed in terms of the latter. In the preliminary draft of the revised code, the heat balance is expressed in terms of " fuel as fired " and " dry fuel," and the heat balance is modified to meet the different combinations of boilers, superheaters, economizers and air heaters, and of solid, liquid and gaseous fuels. See *Preliminary Draft of Revised Code for Stationary Steam-generating Units*, Mech. Engrg., Sept., 1923, p. 548. It is impracticable to assign specific limiting values to a general heat balance because of the wide range in the various influencing factors, such as nature and quality of fuel, type of furnace and grate, rate of driving and the like; but for a rough approximation, Table 18 may be taken as representative practice.

The heat balance in Table 18 refers to boilers in continuous operation and does not include standby losses. (See paragraph 60.)

The calculations of the various items included in the heat balance are best illustrated by a specific example.

Example 17. — Calculate the various heat losses from the following data:

Heat absorbed by the boiler, 76 per cent of the calorific power of the coal as fired.

Analysis of coal as fired:

	Per Cent		Per Cent
Carbon	65	Ash and sulphur	13
Oxygen	8	Free Moisture	8
Hydrogen	5	Nitrogen	1

Calorific value as fired, 11,850 B.t.u.

[1] Rules for Conducting Evaporation Tests of Boilers, A.S.M.E., Code of 1915.

Flue-gas analysis:

	Per Cent		Per Cent
CO_2	12.8	CO	0.6
O_2	5.4	N_2	81.2 (by difference)

Temperature of air entering furnace, 70 deg. fahr.; temperature of flue gases, 470 deg. fahr.; temperature of the steam in the boiler, 340 deg. fahr.; relative humidity of air entering furnace, 80 per cent; combustible in the dry refuse, 20 per cent.

The heat distribution may be referred to the coal as fired, dry coal, or combustible. In this problem it is referred to the coal as fired.

CALCULATION

Solution. — The combustible in the ash, referred to the coal as fired, is $20 \times 13 \div (100 - 20) = 3.25$ per cent or 0.0325 lb. per lb. of coal. Taking this as carbon, the actual weight of carbon burned and appearing in the chimney gas is $0.65 - 0.0325 = 0.6175$ lb. per lb. of coal as fired.

The weight of dry chimney gas per lb. of carbon is (equation 20)

$$G = \frac{4 \times 12.8 + 5.4 + 700}{3\,(12.8 + 0.6)} = 18.82$$

For the carbon actually burned, this is $18.82 \times 0.6175 = 11.62$ lb. per lb. of coal as fired.

The dry air supplied per lb. of carbon burned is (equation 21)

$$A_2 = \frac{3.032 \times 81.2}{12.8 + 0.6} = 18.36.$$

For the carbon actually burned, this is $18.36 \times 0.6175 = 11.34$ lb. per lb. of coal as fired.

DISTRIBUTION OF ACTUAL LOSSES PER POUND OF COAL AS FIRED

Equation	Loss	Calculation	B.t.u.	Per Cent
.....	Heat absorbed by boiler	$0.76 \times 11{,}850$	9,006	76.00
31	Dry chimney gas	$11.62 \times (470 - 70)\ 0.24$	1,115	9.40
40	Incomplete combustion	$0.6175 \times 10{,}160 \times 0.6 \div (12.8 + 0.6)$	280	2.36
41	Combustible in refuse	$0.0325 \times 14{,}600$	474	4.00
33	Moisture in the fuel	$0.08\ [1090.6 + 0.46 \times 470 - 70]$	99	0.83
34	Moisture from combustion of hydrogen	$9 \times 0.05\ [1090.6 + 0.46 \times 470 - 70]$	556	4.70
35	Moisture in the air	$0.8 \times 0.00115 \times 13.2 \times 11.34 \times 0.46\ (470 - 70)$	25	0.20
.....	Radiation and unaccounted for	By difference	295	2.51
	Total		11,850	100.00

TABLE 18

TYPICAL HEAT BALANCE — BITUMINOUS COAL — BASED ON COAL AS FIRED

(No Economizers)

	Excellent Practice	Good Practice	Average Practice	Poor Practice
	Per Cent of Calorific Value of Coal as Fired			
Heat absorbed by the boiler	80.0	75.0	65.0	60.0
Loss due to the evaporation of free moisture in the coal	0 5	0.6	0.6	0.7
Loss due to the evaporation of water formed by the combustion of hydrogen . . .	4.2	4.3	4 3	4 4
Loss due to heat carried away by the dry flue gas	10 0	13 0	17.5	20.0
Loss due to carbon monoxide . .	0 2	0 3	0.5	1.0
Loss due to combustible in the ash and refuse	1.5	2.4	4.5	5.5
Loss due to heating moisture in the air .	0 2	0.2	0.3	0.4
Loss due to unconsumed hydrogen, hydrocarbons, radiation and unaccounted for. . . .	3 4	4 2	7 3	8 0
Calorific value of the coal .	100 0	100.0	100.0	100 0

TABLE 19

TYPICAL HIGH-EFFICIENCY HEAT BALANCE — (NO ECONOMIZERS)

(From Actual Test Results)

Per Cent of Heat Value of Fuel

Kind of Fuel .. .... / Load, per cent rating	Oil 140	Bulk Coal 149	Powdered Coal 143	Natural Gas 140	Lignite 138
Heat absorbed by boiler	82.82	82 4	82 7	82.5	77.0
Free moisture loss. .	0 01	0 6	0.3	0.1	2.0
Hydrogen-moisture loss	6.44	3 8	4 0	9.5	4 5
Air-moisture loss . . .	0.15	0 2	0 1	0.1	0.2
Dry flue-gas loss	7 30	10.0	9 5	6 5	11.0
Loss due to CO. .	0 00	0 3	0 0	0 0	0.0
Combustible in ash	0 00	2.1	0 4	0.0	3.5
Unaccounted for .	3 28	0.6	3 0	1 3	1.8
Total . .	100 00	100 0	100 0	100.0	100.0

59. Inherent Losses. — The heat balance as ordinarily calculated gives the distribution of the **actual** losses. Some of these losses may be considerably reduced or even entirely eliminated, while others are **inherent** and cannot be prevented. A heat balance giving the extent of the inherent losses will show at a glance where improvement may be made and where further gain is impossible. A boiler and furnace may be per-

fect in operation and still fail to utilize the total heat value of the fuel. For example, in the modern boiler (without an economizer or its equivalent) the flue gas cannot be lowered below the temperature of the heating surface with which it was last in contact. Since this temperature corresponds to that of the steam in the boiler, we have as the inherent losses:

1. Heat absorbed by the theoretical weight of dry chimney gases in being heated from boiler room to boiler steam temperature.

2. Heat required to evaporate and superheat the moisture in the fuel from boiler room to boiler steam temperature.

3. Heat required to evaporate and superheat the H_2O formed by the combustion of hydrogen in the fuel from boiler room to boiler steam temperature.

4. Heat required to superheat the moisture in the air (theoretical requirements) from boiler room to boiler steam temperature.

Example 18. — Determine the inherent losses from the data given in Example 17.

Solution. — Proceed as in following chart:

DISTRIBUTION OF INHERENT HEAT LOSSES PER POUND OF COAL AS FIRED

	B.t.u.	Per Cent
1. Inherent loss in the dry chimney gas, $9.26^* \times (340 - 70)\ 0.24$	600.0	5.06
2. Inherent loss due to moisture in coal, $0.08\ (1090.6 - 70 + 0.46 \times 340)$	94.1	0.79
3. Inherent loss due to H_2O formed by the combustion of hydrogen, $9 \times 0.05\ (1090.6 - 70 + 0.46 \times 340)$	529.6	4.47
4. Inherent loss due to "humidity" of the air, $0.8 \times 0.00115 \times 13.3 \times 8.92^* \times 0.46\ (340 - 70)$	13.5	0.11
5. Heat absorbed by ideal boiler (by difference)	10,612.8	89.57
	11,850.0	100.00

* For perfect combustion

A comparison of the actual and inherent losses in percentages of coal as fired is as follows:

	Actual	Inherent
1. Dry chimney gases	9.40	5.06
2. Incomplete combustion	2.36	0.00
3. Combustible in the refuse	4.00	0.00
4. Moisture in the air	0.20	0.11
5. Moisture in the coal	0.83	0.79
6. Moisture due to combustion of hydrogen	4.70	4.47
7. Radiation and unaccounted for	2.51	0.00
8. Heat absorbed by the boiler	76.00	89.57
	100.00	100.00

The difference between the actual and inherent loss is designated as **preventable**. Although the losses due to "incomplete combustion," "combustible in the refuse," and "radiation and unaccounted for" are theoretically preventable, it is almost impossible to entirely eliminate them in practice. The minimum practical loss depends upon the nature of the equipment, grade of fuel, and rate of driving, and must be determined for each installation by actual test. This is also true for the "preventable" loss in the dry chimney gases and that due to the moisture in the air, moisture in the coal, and moisture resulting from the combustion of hydrogen.

Since the ideal or perfect boiler, under the specified conditions, is able to absorb only 89.57 per cent of the calorific value of the coal, it is evident that the actual boiler has a true efficiency of $76 \div 0.8957 = 84.8$ per cent.

If an economizer is used, the inherent losses become less, since the flue gas may be reduced to a temperature considerably lower than that of the steam; but they can never be entirely eliminated unless the flue gas is discharged at a temperature somewhat lower than that of the air entering the furnace.

60. Standby Losses. — The heat balance, as ordinarily calculated, refers only to the heat distribution for continuous operation over a limited period of time. It does not represent average operating conditions, since the various standby losses are not considered. These include: (1) heat lost in shutting down boilers; (2) coal required to start up cold boilers; (3) coal burned in banking fires; and (4) heat discharged to waste in "blowing off" and in cleaning boilers. The magnitude of the standby losses depends upon the size and character of the boiler equipment and the conditions of operation, and may range from 5 to 15 per cent or more of total heat generated (yearly basis). Thus, a continuous 24-hour full-load test may show that 80 per cent of the heat of the coal is absorbed by the boiler, but when the heat represented by a month's evaporation is divided by the heat of the fuel fed to the furnace during the same period, the efficiency may drop to 70 per cent or lower. The standby losses are dependent upon so many variable factors that even average figures may be misleading unless limited to a narrow field of operation. Table 20 gives the heat balance, including standby losses, of the Colfax Station for the year 1920. The data in Table 21, compiled from carefully conducted tests at the central heating and power plant of the Armour Institute of Technology, serve to illustrate the extent and influence of the standby losses on the overall efficiency in a specific case.

Table 22 gives the weight of coal burned in shutting down boilers, in starting up cold boilers, and in banking fires, for a number of Chicago plants.

TABLE 20

TYPICAL HEAT-BALANCE DATA — COLFAX STATION

(Includes Standby Losses)

	October	November	December	January
Coal, as fired, B.t.u. per lb	13,159	13,198	13,283	13,177
Ash as fired, per cent	9 28	9 37	9 14	9 05
Moisture, as fired, per cent	4 45	4 12	3 91	4.49
Inlet air temperature, deg. fahr.	70	70	70	70
Exit gas temperature, deg fahr.	465	467	478	471
Carbon as fired, per cent	72 29	72 49	72 86	72 45
Hydrogen as fired, per cent	4 92	4 93	4 96	4 93
CO_2, per cent	9 5	10.3	11 2	11 1
Combustible in refuse, per cent	26 36	29 10	26 99	26 76
	B t u Per Cent	B t u Per Cent	B t u Per Cent	B t u Per Cent
Heat absorbed by boilers	78 00	77 20	76 8	77 60
Moisture loss	0 42	0 38	0 36	0 42
Hydrogen loss	4 14	4 14	4 16	4 15
Stack loss	13 02	12 01	11 46	11 40
Ashpit loss	3 66	4 23	3 69	3 64
Heat accounted for	99 24	97 96	96 47	97 21
Heat unaccounted for	0 76	2 04	3 53	2 79
Total	100 00	100 00	100 00	100 00

The loss due to **blowing off** depends largely upon the quality of the feedwater. Water containing considerable scale-forming elements requires frequent blowing off, the amount discharged per " blow " varying from 1/2 to 2 gages of water. For example, the 350-hp. Stirling boiler in the power plant of the Armour Institute of Technology (Table 21) is blown off once in 24 hours when in continuous operation, the amount averaging 3 in., as indicated by the water gage. For one month this totals 74,800 lb. The heat lost is 74,800 (338 − 205) = 9,950,000 B.t.u., approximately, or sufficient to evaporate 9500 lb. of water from a feed temperature of 205 deg. fahr. to steam at 100 lb. gage. This amount should be deducted from the water fed to the boiler in calculating the net evaporation (the quality of the steam, of course, being taken into consideration). Compared with the monthly evaporation, the loss in this particular installation is negligible, though it represents an appreciable loss *per se.*

The steam required in blowing soot from the tubes of a return-tubular boiler ranges from 250 to 400 lb. of steam per cleaning with " hand blowing," and from 200 to 350 lb. with mechanically operated " soot blowers." For water-tube boilers the range is considerably greater, depending upon

TABLE 21

INFLUENCE OF STANDBY LOSSES ON OVERALL BOILER AND FURNACE EFFICIENCY

Period Covered by Test	January*	October	July†
Number of hours in month	744	744	744
Hours in service	708	624	153
Hours banked, or out of service	36	120	591
Per cent of rating developed, average for month	133.0	60.2	13.2
Total water:			
Fed to boiler, lb.	11,375,390	5,235,420	791,610
"Blowing off," lb.	74,800	39,870	16,150
Net evaporation	11,366,340	5,230,210	789,990
Total coal:			
Fed to furnace, lb.	1,360,370	728,360	158,960
Burned in banking, etc., lb.	3,680	13,850	37,610
Used for evaporation, lb.	1,356,690	714,510	121,350
Apparent evaporation per lb. of coal fed to furnace, lb.	8.35	7.19	4.98
Actual evaporation per lb. of coal used for evaporation, lb.	8.38	7.32	6.51
Gross overall efficiency of boiler and furnace, per cent	71.9	61.8	44.0
Overall efficiency, deducting standby losses, per cent	72.0	63.2	57.6

* January and October tests. 350-hp. Stirling boiler equipped with chain grate, feedwater 205 deg. fahr., pressure 100 lb. gage, Illinois No 3 washed nut.

† July test 250-hp. ditto

TABLE 22

COAL BURNED DURING BANKING PERIODS*

Rated Capacity of Boiler	Kind of Stoker	Ratio Heating to Grate Surface	Kind of Coal	Hours Banked	Coal Fed to Furnace, Lb. per Boiler Hp.-hr. A	Coal Fed to Furnace, Lb. per Boiler Hp.-hr. B	C
250	Stationary grate	35	Buckwheat	8	0.20	0 35	. . .
500	Chain grate	65	Bit. scrg.	13	0.40	0.52	1000
350	Chain grate	40	Bit. No. 3	9	0.32	0 62	1600
250	Chain grate	48	Bit. scrg.	7	0.35	0 71	1450
1200	Underfeed	82	Bit. scrg.	10	0.18	0.20	2600
550	Underfeed	66	Bit. scrg.	9	0 29	0.37	1165
150	Stationary grate	40	Bit. mine run	12	0.58	0 69	560
75	Stationary grate	48	Poc. lump	12	0 81	0.95	300
400	Murphy	52	Bit. scrg.	13	0.26	0.33	1350

(*A*) Coal fired during banking period.

(*B*) Coal fed to furnace during banking period including that required to put boiler into service at end of banking period.

(*C*) Coal fed to furnace to put cold boiler into service, lb.

* These values are for specific cases only. The range in practice is so wide that average values are of little service for estimating purposes.

the size of the units, number of blowing elements, and the time interval between cleanings. Knowing the number of nozzles, the initial steam pressure, and the time the nozzles are in operation, one may closely approximate the total steam consumption per cleaning from Napier's rule (See equation (280)). A rough approximation is 500 to 750 lb. per cleaning for hand blowing, and 400 to 600 lb. for mechanical blowers incorporated within the setting per 2500 sq. ft. of tube surface.

Tests of Hand and Mechanical Soot Blowers: Report of Prime Movers Committee, N.E.L.A., T.3–22, 1922, p. 47; Power, Aug 26, 1924, p 326.

Keeping Down the Furnace Losses. Power, Mar. 7, 1922.

For a description of the various types of measuring instruments used in calculating heat losses and in establishing heat balances, consult Chapter XVIII.

61. Combustion Control. — For uniformly complete combustion, the fuel and air supply must be correctly proportioned to each other and both must be in proportion to changes in load. While it is possible to obtain this regulation by hand control for a short period of operation, it has been found impractical to effect the desired result in this manner for extended periods. Expert attention can be maintained on a boiler for short runs, but for everyday commercial operation few plants can afford the expense. Mechanical apparatus for automatically controlling the fuel and air supply in proportion to the load demand has been on the market for several years, and while air and fuel adjustments can be more promptly and accurately made by such apparatus than by hand, it cannot take the place of an expert fireman. Without automatic control the fireman is required to make adjustments for each small change in load; with the control in operation these small changes are automatically taken care of and he can devote his attention to irregularities caused by varying quality of coal, clinker formation and the like. Combustion controls are not automatic in the same sense that the governor controls the speed of a turbine; on the contrary, the quantities of air and fuel and the ratios of one to the other must be coordinated to meet the special characteristics of each equipment and operating conditions and must be adjusted from time to time to meet such irregularities as may arise. That properly designed combustion-control apparatus in charge of competent firemen is productive of high efficiency and well worth the extra cost, is evidenced by the increasing number of plants which are adopting this system. This is true not only for large central stations but for hundreds of small isolated plants equipped with mechanical stokers or designed for burning fuel oil, gas, or powdered fuel.

With natural-draft hand-fired installations, automatic control is neces-

sarily limited to regulation of the damper or air supply. With natural-draft stoker-fired installation, both the speed of the stoker-drive and the position of the damper are automatically controlled. In bulk-coal-burning plants equipped with stokers and mechanical-draft fans, the various drives, blast-gates and dampers are individually controlled and at the same time maintained in synchronism by a master control, so that no matter what the rate of supply may be, they are functioning independently to maintain a fixed ratio between the supply of fuel and air. In gas, oil and powdered-fuel installations, there are obviously no stokers, and the particular mechanism feeding the fuel to the furnace is controlled in a suitable manner.

For a description of a number of popular combustion-control systems, see paragraphs 127 and 161.

Combustion Control· Power, Mar. 6, 1923, p. 354; Apr. 29, 1924, p. 676.
Fuel Saving Effected by Combustion Control. Power Plant Engrg., May 1, 1923, p. 473.
Combustion Control for Boilers. Mech Engrg., Oct 1924, p. 590.

PROBLEMS

1. Calculate the dry air requirements for perfect combustion of the coal "as received," using the analyses in Example 1, Chapter II.

2. Required the character and amount by weight of the products of combustion resulting from the perfect combustion with dry air of 1 lb. of the coal designated in Problem 1, Chapter II.

3. Calculate the per cent by volume of CO_2 in the dry flue gas per lb. of coal as fired, data as in Problem 2.

4. A pound of pure carbon is burned with air to CO_2 and CO. If 4 per cent of the carbon is burned to CO and the air supplied was 50 per cent in excess of that required for perfect combustion, required the per cent by volume of CO in the dry flue gas.

5. By-product coke-oven gas having the following analysis is burned completely with theoretical air requirements: Per cent by volume — CO_2, 0.75, CO, 6.00, CH_4, 28.15, H, 53.00, N, 12.10. Calculate the cu. ft. of air required per cu. ft. of gas for perfect combustion and the resulting per cent of CO_2 by volume in the flue gas. Assume air and gas to have the same pressure and temperature.

6. Calculate the weight of air supplied per lb. of coal as fired (analysis as in Problem 1) and the weight of the dry gaseous products of combustion, if the flue gas has the following composition, per cent by volume: CO_2, 13.00, O_2, 5.30, CO, 0.44, N_2, 81.26.

7. Neglecting the influence of S and N_2 in the fuel, show that the per cent of CO_2 by volume in the flue gas for perfect combustion may be expressed in terms of the per cent by weight of the free hydrogen, H', per lb. of carbon

8. Calculate the theoretical temperature of combustion if the coal as fired, analysis as in Problem 1, Chapter II, is completely burned with 50 per cent air excess, initial temperature of air and fuel 60 deg. fahr.

9. If coke breeze containing 85 per cent carbon and 15 per cent ash is completely burned under a boiler with 50 per cent air excess, and the flue-gas temperature is 500 deg. fahr., required the heat loss in the flue gas per lb. of fuel as fired if the temperature of the air supply is 80 deg. fahr.

10. If the flue gas resulting from the combustion of the fuel designated in Problem 12 contains 0.5 per cent CO and 12 per cent CO_2 (by volume), required the loss due to incomplete combustion of the carbon, B.t.u. per lb. of coal as fired.

11. Calculate the heat loss in the refuse if the coal as fired has an ash content of 15 per cent and the combustible in the dry refuse is 20 per cent of the dry refuse. Calorific value of the combustible in the ash, 13,600 B.t.u. per lb.

12. Required the heat lost per lb. of coal as fired in evaporating the moisture from the coal designated in Problem 1, Chapter II, if the temperature of the flue gas is 500 deg. fahr., and that of the boiler room, 80 deg. fahr.

13. If crude oil containing 83 per cent of carbon, 14 per cent of hydrogen and 3 per cent of oxygen is burned under a boiler, required the amount of heat lost per lb. of oil due to the formation of water by the combustion of the hydrogen. Flue-gas temperature, 450 deg. fahr.; temperature of the oil, 120 deg. fahr.

14. The following data were obtained from a boiler evaporation test: Heat absorbed by the boiler, 70 per cent of the calorific value of the coal as fired. Analysis of the coal as fired:

	Per Cent		Per Cent
Carbon	65	Ash and sulphur	12
Oxygen	8	Free moisture	10
Hydrogen	4	Nitrogen	1

Calorific value as fired, 11,300 B.t.u. per lb: combustible in refuse, 13,500 B.t.u. per lb.

Flue-gas analysis:

	Per Cent		Per Cent
CO_2	14.18	CO	1.42
O_2	3.55	N	80.85 (by difference)

Temperature of air entering furnace, 80 deg. fahr.; temperature of the flue gas, 480 deg. fahr.; temperature of the steam in the boiler, 350 deg. fahr.; relative humidity of the air entering the furnace, 70 per cent; combustible in the dry refuse, 20 per cent.

a. Calculate the actual losses in per cent of the coal as fired.

b. Calculate the inherent losses in per cent of the coal as fired.

c. Approximate the extent to which the actual losses may be reduced by careful operation and proper design.

CHAPTER IV

STEAM BOILERS

62. General. — The boiler of to-day is substantially the same as that of five years ago, and in the case of the horizontal return tubular boiler, it may be said to be thoroughly standardized. Such changes as have been made are in the direction of structural alterations to keep pace with the requirements of increased pressure and superheat.

The most notable design changes are in the settings and in the size of units. The principles of efficient and smokeless combustion have been so well proven and are admittedly so fundamental, that engineers now build the boiler plant around the furnace. Adequate combustion space has demanded very high-set boilers, especially with powdered coal as a fuel, and this, by allowing complete combustion, has permitted extremely high efficiencies.

All well-designed boilers, when properly set and similarly fired, are capable of practically the same evaporation per lb. of fuel. The use of stokers with boilers over 350 hp. may be said to be universal. Boiler setting is so tied in with stoker setting that the one cannot be considered without the other. Since stokers are different in their furnace requirements, and the burning of the fuel is a function of the stoker, the engineers for this equipment are generally considered as having the right and responsibility of designing the furnace.

The cost, weight, and space required cannot be accurately judged until the exact design is determined. There is as yet no appearance of standardization in the settings of different boilers, but there is a marked tendency toward a standardization of construction details of the boilers themselves. Many states already have adopted the American Society of Mechanical Engineers' "Standard Specifications for the Construction of Steam Boilers and Other Pressure Vessels."[1] Every power plant owner and operator should be conversant with the boiler code, insurance and inspection laws, in the communities where they are in force, and no boiler may be constructed or operated without complying with the requirements of the law.

Regardless of improvements in furnace design, skill, good judgment,

[1] Copies of the Code may be obtained from the office of the Secretary of the A.S.M.E., 27 W. 39th St., New York City.

and continued vigilance are required on the part of the operator to secure good efficiency.

No attempt will be made to analyze the boiler from the standpoint of manufacture, and reference is made to the A.S.M.E. Boiler Code and to current trade catalogues for this phase of the subject. For an excellent treatise on circulation in various types of boilers, consult "The Kidwell Two-flow Ring-circuit Water Tube Boiler" by Edgar Kidwell, and published by the Kidwell Boiler Co., Milwaukee, Wis.

A general classification of steam boilers is unsatisfactory because of the overlapping of the various groups. They may be classified according to (1) method of firing, as **externally** and **internally** fired; (2) relative position of the heated gases and water, as **water-tube** and **fire-tube**; (3) arrangement of tubes, as **vertical, horizontal,** and **inclined**; (4) curvature of the tubes, as **straight tube** and **curved tube**; (5) nature of service, as **stationary, marine,** and **locomotive**; (6) direction of the gases, as **through-tube,** and **return-tubular**; (7) baffling, as **horizontal-baffle,** and **vertical-baffle**; (8) steam pressure, as **high-pressure** and **low-pressure**; (9) location of the drum, as **longitudinal-drum** and **cross-drum**; and so on. A few popular types will be described in detail.

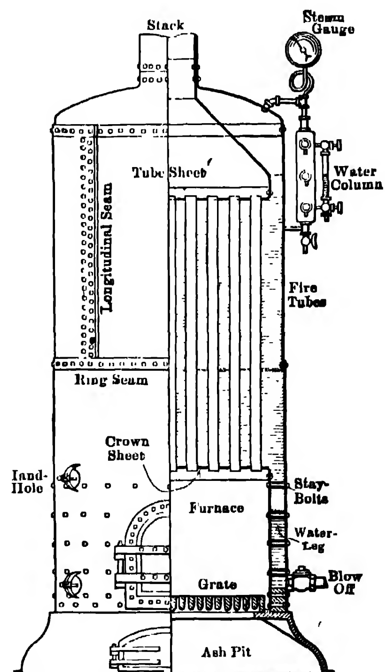

FIG. 19. Vertical Tubular Boiler with Submerged Tube Sheet.

63. Vertical Tubular Boilers. — Figures 1 and 19 illustrate typical portable fire-tube boilers of the **internally-fired type.** They are used only where small power, compactness, low first cost, and semi-portability are the chief requirements. Boilers of this type have a cylindrical shell with a fire box in the lower end and with tubes running from the crown sheet of the furnace to the upper tube sheet at the top of the boiler. They are built in various sizes ranging from 24 to 48 in. in diameter, and from 60 to 120 in. in length, with corresponding heating surface of 50 to 500 sq. ft. (5 to 50 hp.). The tubes are usually 2 in. in diameter and the working pressure seldom exceeds 100 lb. per sq. in. gage.

The tubes are placed symmetrically with a continuous clear space between them and these spaces cross the tube section at right angles. This arrangement permits the tubes and tube sheet to be readily cleaned. Two styles are in common use — the **exposed-tube,** Fig. 1, and the **submerged-tube,** Fig. 19. In the former, the tube sheet and the upper portion of the tubes are exposed to the steam and in the latter they are completely submerged. According to the A.S.M.E. Boiler Code, not less than seven handholes or wash-out plugs are required for boilers of the exposed-tube type; three in the shell at or about the line of the crown sheet, one in the shell at or about the fusible plug, and three in the shell at the lower part of the **water leg.** In the submerged type, two or more additional hand holes are required in the shell in line with the upper tube sheet. The distance between the crown sheet and the top of the grate should never be less than 24 in. even in the smallest boiler, and should be as great as possible, to insure good combustion.

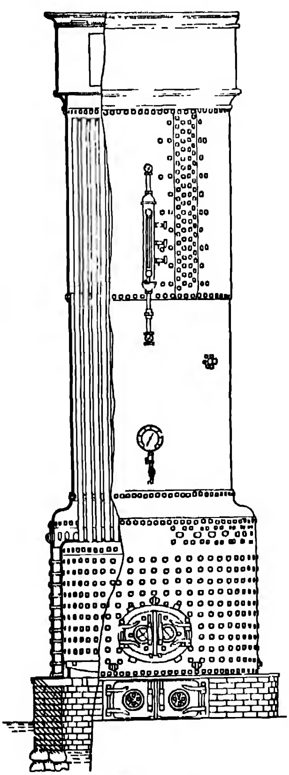

FIG. 20. Manning Boiler.

In some designs the furnaces are constructed of corrugated steel, thus doing away with the stay bolts.

The advantages of this type of boiler are as follows: (1) it is compact and portable; (2) it requires no setting beyond a light foundation; (3) it is a rapid steamer; and (4) it is low in first cost. It has the following disadvantages: (1) it is not easily accessible for thorough inspection and cleaning; (2) the steam space is small, resulting in excessive priming at heavy loads; (3) the economy is poor, except at light loads, as the products of combustion escape at a high temperature on account of the shortness of the tubes; (4) smokeless combustion is practically impossible with bituminous coals; (5) the small water capacity results in rapidly fluctuating steam pressures with varying demands for steam.

Although vertical fire-tube boilers of the portable or semi-portable type are seldom constructed in sizes containing more than 500 sq. ft. of heating surface, other types of vertical fire-tube boilers, of which the **Manning** (Fig. 20) is a well-known example, are not limited to small sizes and have been constructed with heating surface of 6000 sq. ft. per unit. Many of the disadvantages found in the smaller types are obviated in the

Manning boilers, which, as far as safety and efficiency are concerned, rank with any of the other first-class types. They differ from the boiler described above mainly in having the lower or furnace portion of much greater diameter than the upper part which encircles the tubes. This permits a proper proportion of grate, which is not obtainable in boilers like those shown in Figs. 1 and 19. The double-flanged head connecting the upper and lower shells allows sufficient flexibility between the top and bottom tube sheets to provide for unequal expansion of tubes and shell. The ashpit is built of brick and the water leg does not extend below the grate level, thus doing away with dead-water space. Where overhead room permits and ground space is expensive, this boiler offers the advantage of taking up a small floor space as compared with horizontal types. The Manning type of boilers is designed for steam pressures up to 200 lb. gage.

64. Locomotive-type Boiler. — This style of boiler is used occasionally for stationary power service where semi-portability is desired, as in connection with agricultural and saw-mill plants. It is also used to a limited extent for low-pressure heating work. It is in basic principle a vertical, internally fired boiler placed horizontally. Two general designs are in common use: (1) the **water-bottom,** in which the fire box is entirely surrounded by water (Fig. 21); and (2) the **open-bottom** (Fig. 22) in which the fire box is submerged on all sides but not on the bottom. Water-bottom boilers are self-contained and require no settings whatever, but the open-bottom boilers, particularly those fitted with enlarged fire boxes, frequently require a masonry ashpit. These boilers are available in standard sizes ranging from 150 to 2500 sq. ft. of heating surface, but larger units have been built for special purposes. Working pressures range from 15 lb. to 150 lb. gage. For anthracite coal, the tubes range from

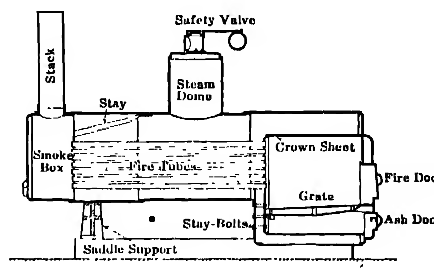

FIG. 21. Locomotive Type — Water Bottom.

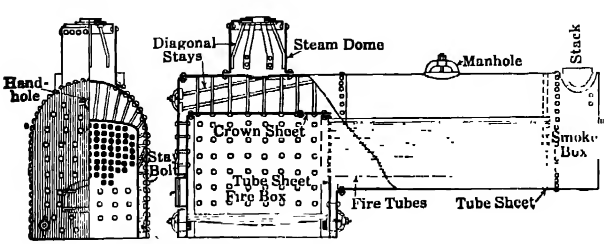

FIG. 22. Locomotive Type — Dry Bottom.

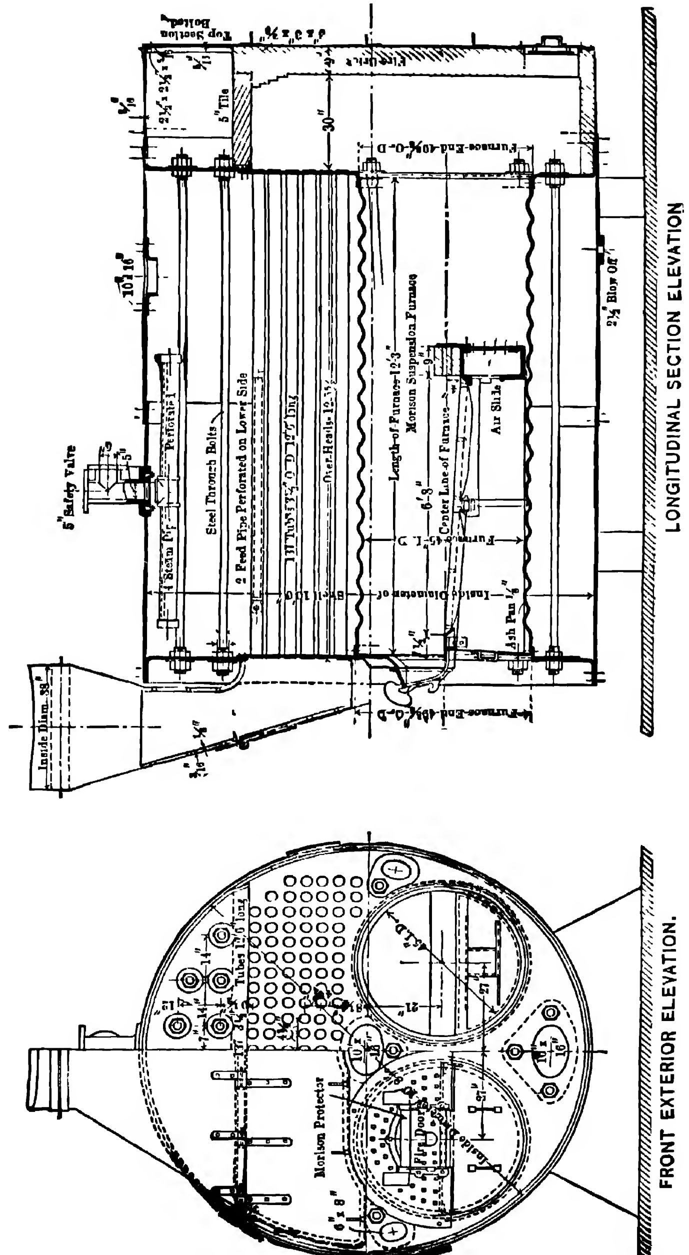

FIG. 23. 250 Hp. Stationary Scotch-marine Boiler

2 to 3 in. in diameter, and for bituminous coals from 3 to 4 in. Sizes with less than 300 sq. ft. of heating surface per hp. are ordinarily furnished with a dome located over the fire box, while the larger sizes are constructed with either dome or dry pipe. This style of boiler is not much in evidence in high-pressure stationary plants.

65. Stationary Scotch Marine-type Boilers. — These boilers belong to the internally-fired return-tubular type; they are self-contained, require no brick setting, occupy little overhead room, and are excellent steamers. The shell is cylindrical, fitted near the bottom with one or more cylindrical furnaces traversing the entire length of the shell, and partly filled above the furnaces with full-length return tubes. The gaseous products of combustion are guided from the furnaces to the return tubes by a back-connection or combustion chamber. The furnace and tubes are entirely surrounded by water, so that all fire surfaces, excepting the rear of the combustion chamber, are water cooled. Figure 23 shows a section through a popular design, in which the furnace is corrugated. These corrugations, in addition to giving greater strength to the furnace, act as a series of expansion joints, taking up the strains due to unequal expansion of furnace and shell. In other designs, the furnace is strengthened by the **Adamson Ring,** or **Bolling Hoop.** In the former, the furnace sections are flanged outward and riveted together through a ring inserted between them (Fig. 24), while in the latter, the sections are riveted to special expansion joints (Fig. 25). The single furnace boiler is constructed in sizes ranging from a small unit 35 in. in diameter by 52 in. in length (60 sq. ft. of heating surface) and rated at 6 hp. to units 96 in. in diameter by 17 ft. in length (1500 sq. ft. of heating surface) and rated at 150 hp. Stationary boilers with two furnaces have been constructed with shells 120 in. in diameter, and 20 ft. in length, and rated at 300 hp. For marine service, this type of boiler has been built with as many as four furnaces and with shell diameters up to 20 ft. and overall length of 11 ft., the size being limited only by transportation facilities. For stationary purposes this type of boiler is designed for working steam pressures ranging from 100 to 200 lb. per sq. in., and for sizes up to 300 hp. While a number of these boilers are to be found in stationary steam plants where low head room is essential, such as in office buildings, large apartment buildings, and hotels, they play a relatively unimportant part in steam generation for power purposes. The normal circulation in the standard Scotch marine-type boiler is defective, because

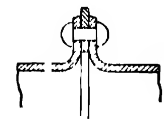

Fig. 24. Adamson Ring.

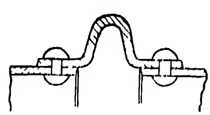

Fig. 25. Bolling Hoop.

the water lies "dead" in the bottom of the shell, and the unequal expansion and contraction of furnace walls and tubes tends to cause considerable tube leakage.

Figure 26 shows a section through a modified type (suitable for low-volatile coals) which facilitates circulation of the water below the furnace. The tubes are in two nests, the usual return tubes and a number of short ones leading from the rear head of the furnace to the combustion chamber. The object of the short tubes is to divert the flame downward and to heat the rear and lower portions of the boiler. This increases the rate of circulation.

The advantages of Scotch boilers and of most internally fired boilers are: (1) low head room; (2) minimum radiation losses; (3) no setting required; (4) no leakage of cold air into the furnace, as frequently occurs

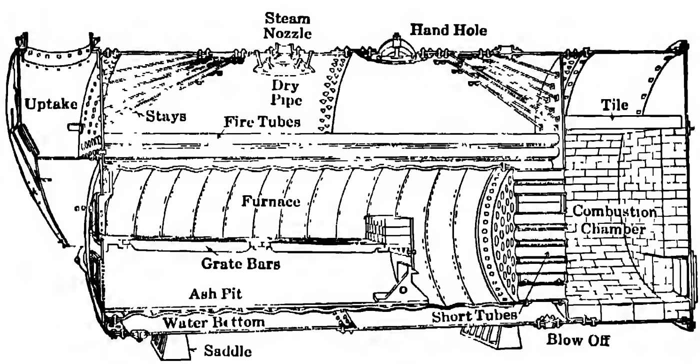

FIG. 26. "Duplex" Internally Fired Boiler.

through cracks or porous brickwork of other types, and (5) large steam capacity for the space occupied. With high-volatile coals, the furnace volume is insufficient for efficient and smokeless combustion at high ratings, and this fact, together with the limitation in sizes, due to transportation facilities, precludes the use of this type of unit in large central stations. Extension furnaces, hand- or stoker-fired, may offset the limitations of the small internal furnace, but this addition neutralizes the chief advantage of the internally fired type, namely, compactness and absence of masonry setting.

The **Cornish, Lancashire,** and **Galloway** boilers are common in Europe, but are seldom found in American practice. The Cornish boiler is essentially a single-flue Scotch unit without return tubes and is the oldest design among modern internally fired boilers. The Lancashire is an improvement on the Cornish boiler in that there are two flues instead of one. In the Galloway boilers the two furnace flues merge beyond the

bridgewall into one large flue which is traversed radially throughout its length by conical water tubes. These boilers are set in brickwork, so arranged that the gases, after leaving the furnace, pass forward below the outer shell and then backward along the sides of the shell to the uptake.

66. Horizontal Return-tubular Boiler. — These boilers are the most widely distributed steam generators in the United States and may be regarded as the standard American type. They owe their popularity to low first cost, high evaporative capacity, compactness, and low overhead space requirements. Standard sizes range from a small unit 36 in. in diameter by 8 ft. in length (150 sq. ft. of heating surface) and rated at 15 hp. to a unit 84 in. in diameter by 20 ft. in length (3500 sq. ft. of heating surface) and rated at 350 hp. There is no particular limit to the

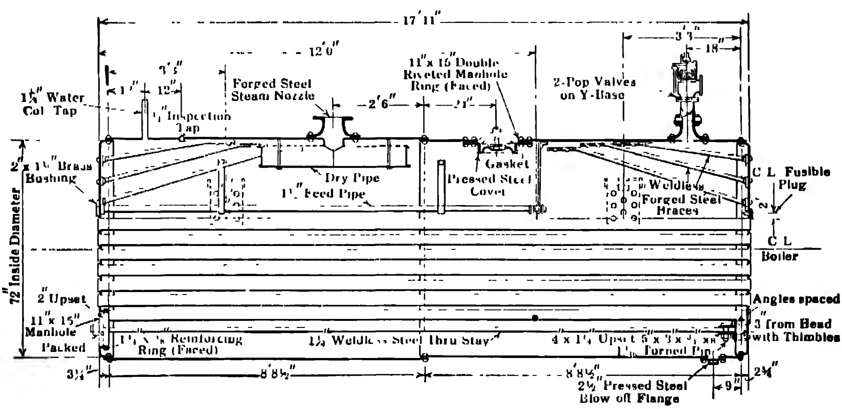

FIG. 27. Longitudinal Section through a 150 Hp. Horizontal Return-tubular Boiler.

sizes of these types except shipping facilities, and a few units have been built as large as 108 in. in diameter and 21 ft. long and rated at 500 hp.; but as a general rule some other type is selected where the desired rating exceeds 250 hp. They are usually designed to operate at pressures varying from 10 to 150 lb. per sq. in., but there are a number of special designs operating at 175 lb. Figure 27 shows a longitudinal section through the shell of a popular make of return-tubular boiler, and Fig. 28 gives a perspective view of another design with extended shell. The drawings are self-explanatory. The tubes are usually 3, 3 1/2 or 4 in. in diameter, the smaller tubes for low-volatile fuels, and the larger tubes for high-volatile fuels. As a general rule. 4-in. tubes are not furnished with shells under 48 in. in diameter. Boilers over 54 in. in diameter are usually fitted with a **dry pipe** (Fig. 27), for separating moisture from the steam,

while the very small sizes have a **fixed dome** (Fig. 29), flanged at the base and riveted to the shell, or in special cases, an **independent dome** at-

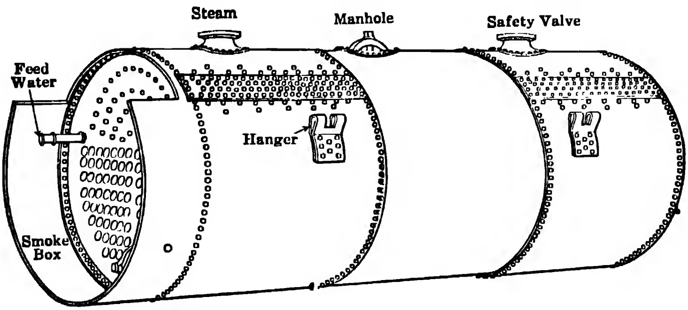

FIG. 28. Horizontal Return-tubular Boiler — Extended Shell.

tached to the shell with a nipple or a nozzled connection. The requirements for plate thickness, riveted joints, staying, supports, handholes, manholes, and other construction details are fully specified in the A.S.M.E., state, and insurance codes.

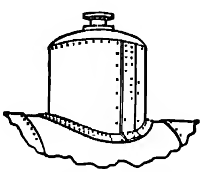

FIG. 29. Steam Dome.

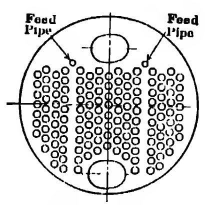

FIG. 29a. Tube Arrangement—"Uniflow" Boiler.

Return-tubular boilers are made either with an **extended** or **half arch front,** Fig. 30, or **flush front,** Fig. 116. The shell may be supported by lugs resting on the brickwork, or by steel beams and hangers, Fig. 30. According to the A.S.M.E. Boiler Code, all horizontal tubular boilers over 78 in. in diameter are required to be supported by the **outside-suspension** or **gallows-frame** type of setting. With the side bracket support, the front lugs usually rest directly on iron or steel plates embedded in the brickwork, and the back lugs on rollers to permit free expansion and contraction. The brackets are long enough to rest upon the outside wall, so that the inside brick lining can be removed without disturbing the setting.

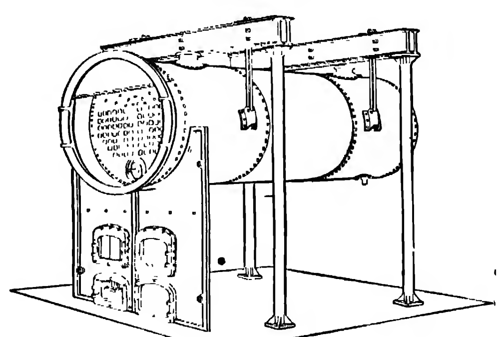

FIG. 30. Horizontal Return-tubular Boiler. Gallows-frame Suspension.

It is asserted that the rate of circulation, and hence the overload capacity, of the return-tubular boiler may be greatly increased by using smaller diameter tubes and grouping them in three sections, as shown in Fig. 29*a*, instead of the uniform vertical spacing of the standard type. In another design the tubes are arranged as in Fig. 28, but the groups are separated from each other by a vertical steel baffle running the entire length of the shell.

Return-tubular boilers are of the externally fired type, and therefore must be provided with a furnace and setting. For a description of the latter, see paragraphs 102 to 105.

A return-tubular fire-box boiler of the portable type, as illustrated in Fig. 31, is finding favor with many engineers where a compact moderate capacity and self-contained unit is desired. As will be seen from the cut, the front of the boiler is cylindrical in form and extends over the furnace, while the rear end is oval, the lower portion extending below the cylindrical part far enough to hold the short tubes leading from the furnace to the back connection. The products of combustion, passing through the short tubes and into the back connection, are carried by the return tubes through the upper section of the boiler to the stack. The front and sides of the furnace are lined with fire brick. These boilers are available in standard sizes ranging from 20 to 150 hp. and are intended for pressures not exceeding 100 lb. per sq. in.

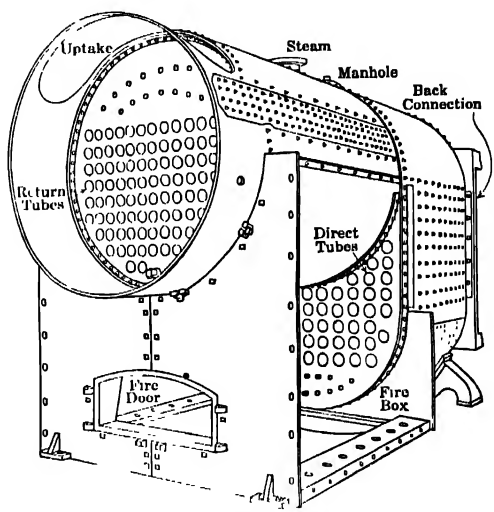

FIG. 31. Return Tubular Portable Fire-box Boiler. (Side Plates Removed to Show Furnace Construction.)

67. Horizontal Water-tube Boilers, Longitudinal Drum. — Figure 32 gives a general assembly of a standard longitudinal-drum **Babcock and Wilcox** boiler, illustrating one of the best known and most widely distributed water-tube boilers in the United States. This particular type is made in single units ranging from 750 sq. ft. of heating surface (75 hp. rated capacity), to 8000 sq. ft. of heating surface (800 hp. rated capacity). The distinguishing features of this boiler are: (1) horizontal drum or drums; (2) inclined or vertical sectional headers, and (3) inclined straight tubes.

The tubes, usually 4 in. in diameter and 18 to 20 ft. in length, are arranged in vertical and horizontal rows and are expanded into cast-iron or pressed-steel headers. Two vertical rows are fitted to each header and are "staggered," as shown in Fig. 33. The headers are connected with the steam drum by short tubes expanded into a **cross box,** Fig. 34, which in turn is riveted to the drum. The headers are either vertical, Fig. 32, or inclined, Fig. 35. Each tube is accessible through individual handhole openings. These openings are elliptical in shape in the vertical headers, because of the inclination of the tubes. This shape is necessary to provide for the insertion and removal of the tubes. Circular **handholes** are ordinarily used in the **inclined headers**

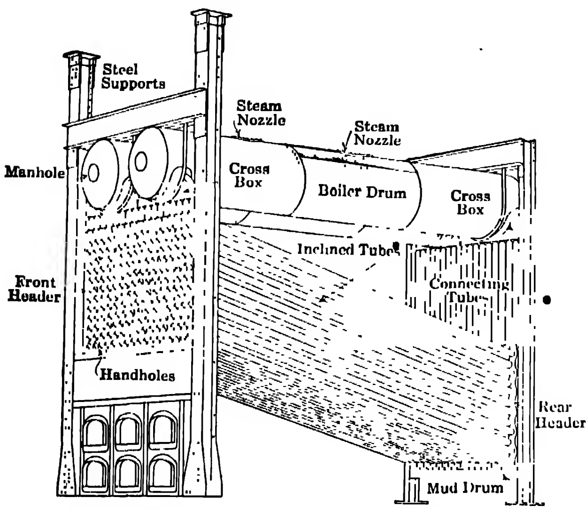

FIG. 32. Babcock and Wilcox Boiler Assembly — Longitudinal Drum Type. (Vertical Header.)

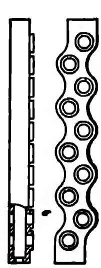

FIG. 33. Details of Header — B. & W. Boiler. (Inclined Header.)

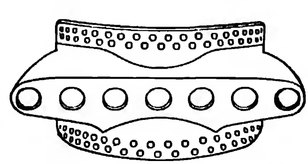

FIG. 34. Cross Box — B. & W. Boiler.

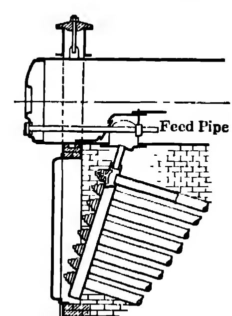

FIG. 35. Front Section — B. & W. Boiler.

where the tubes enter the latter at right angles. The elliptical openings are closed by inside fitting forged covers held in position by steel clamps and bolts, Fig. 36. The circular openings are closed on the outside

by forged steel caps, milled and ground and held in place by clamps and bolts, Fig. 37. Thin gaskets are required with the inside elliptical covers, but not with the outside circular plates. (The main tubes are inclined at an angle of about 22 deg. with the horizontal. The rear

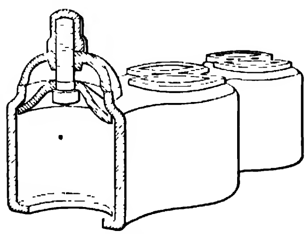

FIG. 36. Elliptical Handhole.

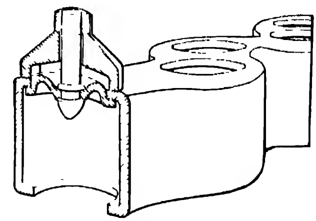

FIG. 37. Circular Handhole.

headers are connected at the bottom to a rectangular forged steel mud drum, by means of nipples expanded into counterbored seats.) The boiler is supported by steel girders resting on suitable columns independent of the brick setting. The feedwater enters the front of the steam drum, as shown in Fig. 35. Circulation is effected by the difference in density between the solid column of water in the rear header and the mixed steam and water in the front one. The longitudinal-drum type of B. & W. boilers under 325 hp. have but one steam drum, and the larger sizes have two or three, depending upon the width of the setting. While a few special designs of this type of boiler have been made for steam pressures as high as 500 lb. per sq. in., in the great majority of power plant installations, the working pressure seldom exceeds 275 lb. per sq. in.

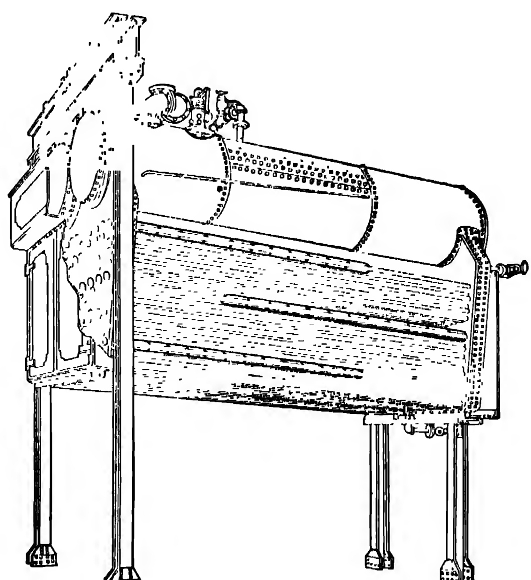

FIG. 38. Heine Boiler — Longitudinal-drum Type.

All water-tube boilers are constructed on the sectional principle; that is, they may be shipped in sections and erected at the power plant. This

removes any restriction on the size of single units that might otherwise be imposed by transportation limitations.

Figure 38 gives a general assembly of a **Heine** longitudinal-drum boiler, illustrating another well-known type of horizontal water-tube boiler. This boiler differs from the Babcock and Wilcox in that the drum is inclined and parallel to the tubes, and the latter are expanded into a single fabricated water leg or header, Fig. 39. The feed-water enters at the front of the steam drum, and flows into the mud drum, from which it passes to the rear header. Steam is taken from the front of the steam drum and is partially freed from moisture by the dry pipe. A baffle over the front header prevents an excess of water from being carried into the dry pipe. As the rear header forms one large chamber, no additional mud drum is necessary. Because of the greater area through the header, the circulation is asserted to be freer than in the longitudinal-drum type of sectional header. Heine boilers are usually fitted with the **"Key"** safety handhole cap, Fig. 40, which requires no yoke, bolt, or gasket. The caps are slipped into place from the inside of the water leg and are held in place by the water or steam pressure.

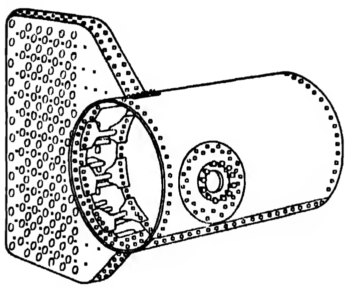

FIG. 39. Junction at Header and Drum — Heine Boiler.

Among other well-known makes of longitudinal-drum, horizontal water-tube boilers may be mentioned the **Edge Moor, Keeler, Parker, O'Brien, Erie City, Kroeschell** and **Casey-Hedges.**

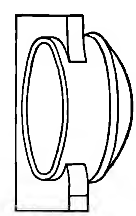

FIG. 40. "Key" Handhole and Cap.

68. Horizontal Water-tube Boilers — Cross-drum Type. — This type of horizontal water-tube boiler has practically superseded the longitudinal type in modern power houses where large units are desired. Among the better known designs may be mentioned the **Springfield, Babcock & Wilcox, Heine "S-type," Wickes, Keeler** and **Page.** As will be seen from Figs. 41 and 42, the cross-drum type differs from the longitudinal-drum chiefly in the location of the drum and the method of support. The drum is placed transversely across the rear, immediately above the rear header or at some point between the top of the headers. Connection between drum and headers is made by means of circulating tubes expanded into bored seats, and extending the full width of the drum. Suitable baffles prevent the water and steam (in the circulating tubes) from discharging openly into the drum. This type of boiler has been built in various sizes ranging from 1200 up to 20,000 sq. ft. of heating surface

(120 to 2000 hp. nominal rating) with working pressures ranging from 100 to 1200 lb. per sq. in.

The headers in the majority of cross-drum boilers have one handhole for each tube end, but in the Springfield design one handhole covers four tubes.

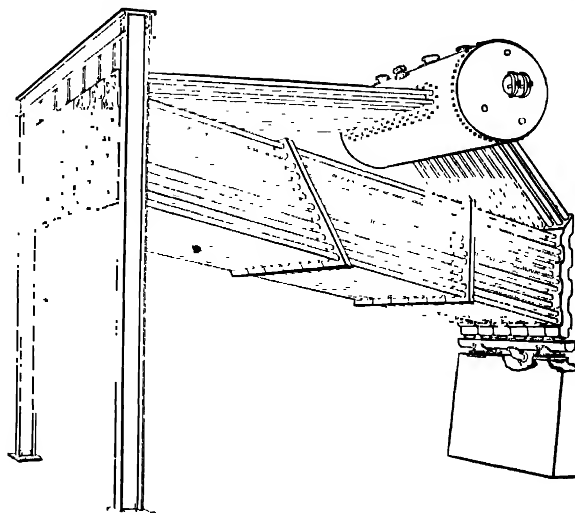

FIG. 41. Springfield Boiler — Cross-drum Type.

Figure 42 shows a special application of the cross-drum design to very high pressures and temperatures. The particular unit illustrated has inclined headers, and the water-heating surface is divided into two decks or sections. The lower deck has 8 sections and the upper deck 17 sections of 2-in. tubes. The lower deck is not baffled, and the upper deck has a vertical baffle, causing the gases to make two passes. A primary and secondary superheater are placed between the two decks of water tubes.

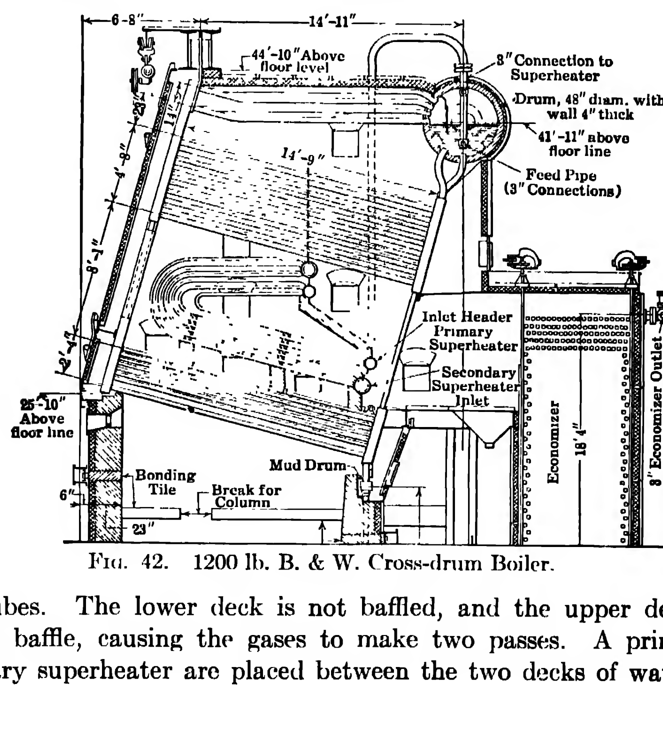

FIG. 42. 1200 lb. B. & W. Cross-drum Boiler.

Two 3 1/4-in. horizontal circulating tubes are connected to the top of each manifold header and the drum, but the circulators from each alternate header are bent downward and sidewise so that they are connected to the drum in the same circumferential row as the circulators without bends. The cross drum is a forged steel cylinder 48 in. in diameter and 4 in. thick, with integral drum heads. The headers have 1 1/4-in. thickness front and back and 5/8-in. sides, and are designed to give the tubes a stagger of nearly 4 in. The mud drum is 7 1/4 in. square, 1 in. thick, and extends through each side of the setting. The primary superheater is designed to raise the temperature of the steam under 1200 lb. pressure to 750 deg. fahr., and the secondary superheater incloses the primary superheater and is intended to raise the temperature of the exhaust from the extra-high pressure turbine to 750 deg. The complete unit is about 28 ft. wide, 36 1/2 ft. deep and 45 ft. above the floor.

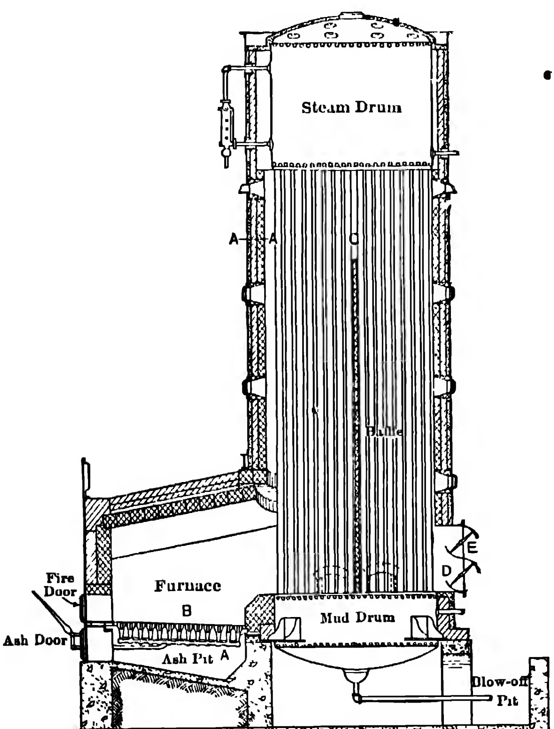

FIG. 43. Wickes Water-tube Boiler — Vertical Type.

Babcock and Wilcox cross-drum boilers have been installed in the Colfax station of the Duquesne Light Co., Calumet and Crawford stations of the Commonwealth Edison Co., Springfield power station of the West Penn. Power Co., and Riverside station of the Northern States Power Co. Notable installations of Springfield cross-drum boilers are in the new Hell Gate station of the United Electric Light and Power Co., Barbados plant of the Counties Gas and Electric Co. and the Indiana Harbor plant of the Inland Steel Co. The units of the Hell Gate station have 18,900 sq. ft. of heating surface with 3-in. tubes, grouped as illustrated in Fig. 89.

69. Vertical Water-tube Boilers, Straight-tube Type. — Figure 43 shows a sectional elevation through a **Wickes** vertical boiler and setting illustrating a well-known design of boiler with vertical, straight water tubes and vertical drums. The steam drum and water drum are arranged one directly above the other. The tubes are expanded and rolled into both tube sheets and are divided into two sections by fire-brick tile. The water line in the upper drum is carried more than 2 ft. above the tube sheet, leaving a space of 5 ft. between the water line and top of the drum. This affords a large steam space and disengagement surface. Feedwater is introduced into the steam drum below the water line and flows downward through the tubes of the second compartment. The boiler is supported by four brackets riveted to the shell of the bottom drum and is independent of the setting. The entire boiler is enclosed in a brick or steel casing insulated with non-conducting material and lined with fire brick. The entire boiler is surrounded by the products of combustion. This type of boiler is simple in design, easy to inspect and clean, low in first cost and takes up little floor space; but, having only two passes, it cannot be forced efficiently to very high ratings. Wickes vertical boilers are built in all sizes up to 5000 sq. ft. of heating surface, and for working pressures varying from 100 to 200 lb. per sq. in.

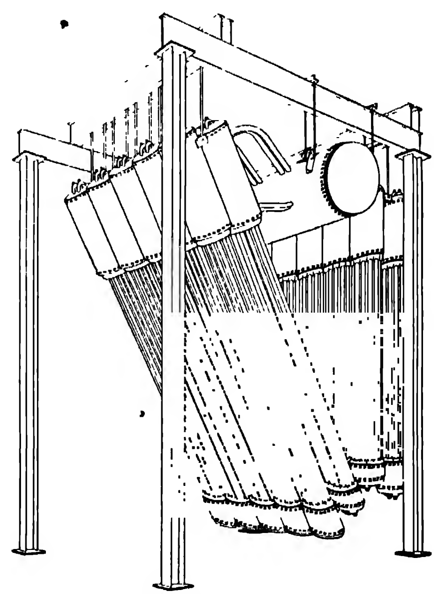

FIG. 44. Bigelow-Hornsby Boiler.

The **Bigelow-Hornsby** boiler, Fig. 44, consists of a number of cylindrical elements, each element consisting of an upper and a lower drum connected by straight tubes. The two front elements are inclined over the furnace at an angle of about 68 deg. with the horizontal, and the two rear elements are vertical. The upper drums of the elements are connected to a horizontal steam drum by flexible tubing, as indicated. Four elements constitute a section, and any number of sections may be connected together into units ranging from 2500 to 15,000 sq. ft. of heating surface. Feedwater enters the top drum of the rear elements and passes the rear tubes and then up the tubes in the front elements. A notable installa-

tion of Bigelow-Hornsby boilers is in the new South Meadow Station of the Hartford Electric Light Co. The units in this station have approximately 13,920 sq. ft. of water heating, and are equipped with integral economizers in the rear of the setting. Each unit is composed of 55 cylindrical elements, 11 of these elements comprising the economizer.

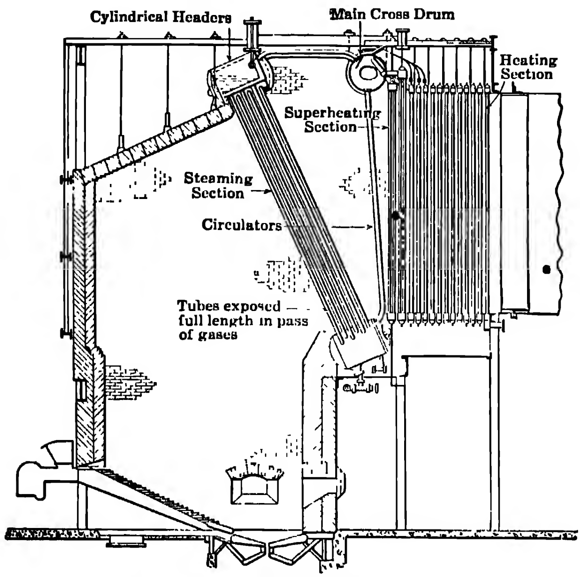

Fig. 45. Edge Moor Single-pass Boiler.

Figure 45 shows a side sectional elevation of a single-pass Edge Moor boiler comprising a number of straight tube cylindrical units and a cross drum connected by slightly curved circulating tubes. It is claimed by the manufacturers that the draft loss is less than half that usually encountered for equal capacity and efficiency, and that the boiler responds rapidly to high rates of evaporation without the usual falling off in efficiency. The best efficiency is obtained when the boiler is evaporating at a rate of about 9 lb. of water per sq. ft. of heating surface, or in other words at 300 per cent rating. At 400 per cent of rating, the results are about equal to those heretofore obtainable around 165 per cent of rating.

70. Vertical Water-tube Boilers — Curved-tube Type. — Under this general heading may be grouped such well-known boilers as the **Stirling** (Fig. 46), **Kidwell** (Fig. 224), **Adams, Heine V-type, Badenhausen** (Fig. 47), **Connelly, Ladd** (Fig. 90), **Erie City Vertical** (Fig. 48), and **Rust.** In all these boilers the drums are horizontal, but the tubes vary in inclination from the vertical, in the Rust, to almost horizontal in the second pass of the Badenhausen.

The standard type of Stirling boiler, Fig. 46, consists of three transverse steam and water drums set parallel and connected to a mud drum by three banks of tubes so curved as to enter the drums radially. The center drum is connected to the front and rear drums by steam-circulating tubes and to the front drum by water-circulating tubes. Steam is

taken from the rear drum. The boiler is suspended on a steel framework entirely independent of the brickwork setting. The feedwater enters the rear upper drum, which is the coolest part of the boiler, and flows to the bottom or mud drum and thence up the front bank of tubes to the front drum, across to the middle drum and finally down the middle bank of tubes to the mud drum. The interior of the drums is accessible for cleaning and inspection by manholes located in the ends. In the "W" type of Stirling boiler, there are three horizontal steam drums and two lower or mud drums. The inclination of the four banks of tubes is such as to form the letter "W." This arrangement exposes a large tube surface to direct radiation. Notable installations of the "W" type Stirling boilers are in the Marysville and Trenton Channel stations of the Detroit Edison Co. Individual boiler units in the Marysville station have 28,212 sq. ft. of effective water-heating surface and are served by a double-ended 22-tuyere, 3-row Taylor stoker with 14 retorts at each face of the boiler.

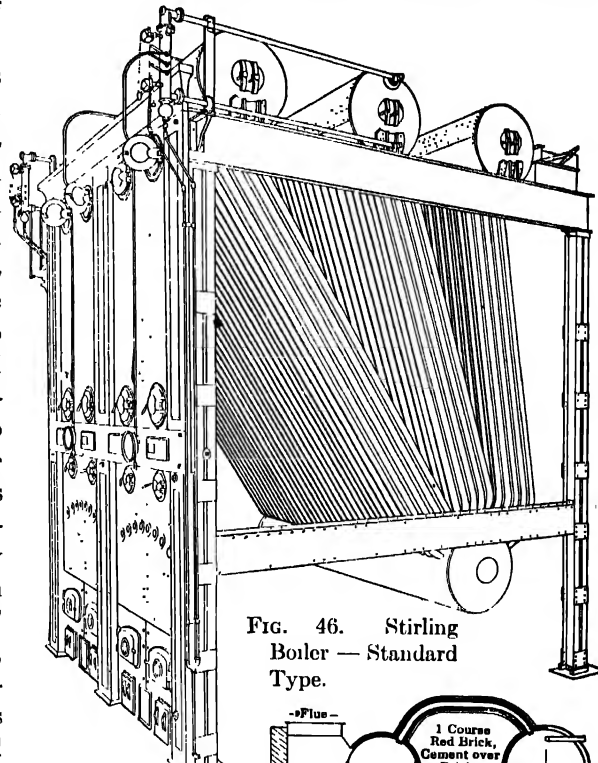

FIG. 46. Stirling Boiler — Standard Type.

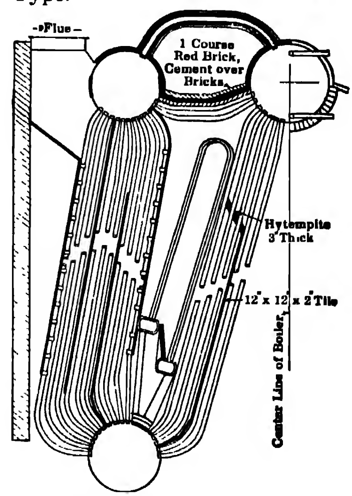

FIG. 46a. Half Section of "W" Type Stirling Boiler — Marysville Station.

The Badenhausen boiler, Fig. 47, and Kidwell boiler, Fig. 224 (which is similar to the former in basic principle) are examples of the "ring-flow" type, in which the circulation is continuous

and unrestricted at all loads. This excellent circulation is due to the fact that the areas of the tubes entering and leaving the drums are practically the same. Feed-water enters the top rear drum and passes down the rear outer tubes to the mud drum, thence up the front bank of the lower front drum, thence across the upper bank of tubes to the top rear drum. The upper front drum is essentially a steam collector, and the tubes connecting the top of this drum with that of the rear top drum are practically superheaters. A notable installation of Badenhausen boilers is in the Highland Park plant of the Ford Motor Co. The boilers are of the preheater type (economizer element integral with the boiler), each unit comprising 25,000 sq. ft. of effective heating surface.

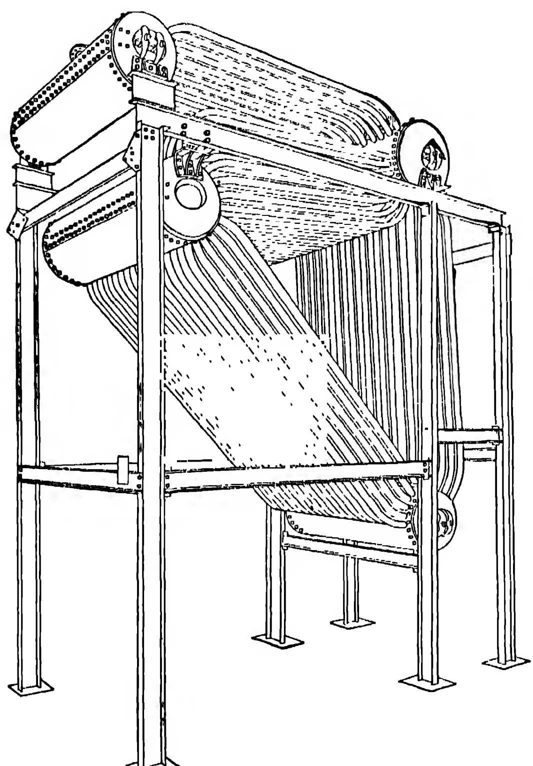

FIG. 47. Badenhausen Boiler.

The Erie City vertical, Fig. 48, and the "standard" Ladd boilers have but two drums and three banks of tubes. The tubes in the former are 3 in. in diameter and in the latter 3 1/4 in. The two horizontal drums are in line vertically and the tubes are so curved at the ends as to enter the drums radially. The boiler proper, comprising the two drums and connecting tubes, is suspended from the upper drum by a framework of steel beams. Feedwater enters the lower drum of the Erie boiler through a distributing compartment, passes up the front bank of tubes and then down the middle and rear banks to the lower drum. In the Ladd boiler, the feedwater enters the separate compartment in the lower drum and is directed upward through the rear bank of tubes, down the middle bank and thence upward in the front bank. The very large Ladd boilers are of the double-ended type and consist essentially of two "standard"

units inclined so as to form an inverted V. One of the most notable installations of the double-ended type is in the River Rouge plant of the Ford Motor Co., Fig. 90. The four main drums of each unit, 60 in. in diameter by 21 ft. in length, are connected by banks of tubes averaging 20 ft. 6 in. in length, giving a total effective heating surface of 26,470 sq. ft. Each unit occupies 29 by 31 ft. of floor space, and the distance from ash floor to the top of the superheating pipe is 83 ft. The furnaces are designed to burn blast-furnace gas and powdered coal, simultaneously and in any ratio.

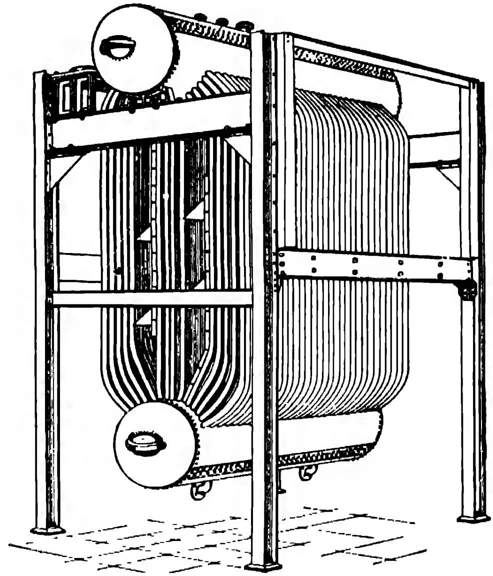

FIG. 48. Erie City Vertical Boiler.

71. Combined Fire- and Water-tube Boilers. — Figure 49 shows a general assembly of a **Kroeschell Fire- and Water-tube** boiler illustrating a combination of a return-tubular boiler with water-tube elements which has many advantages over the plain fire-tube type. This combination has the large storage capacity of the return-tubular boiler and the thorough circulation and rapid steaming property of the water-tube boiler. The water-tube elements, immediately above the fire, prevent the shell from becoming overheated, and the extra pass of the gases permits of lower flue-gas temperature for a given rating. This style of boiler is made in standard sizes, ranging from a 48-in. by 16-ft. unit containing 1040 sq. ft. of

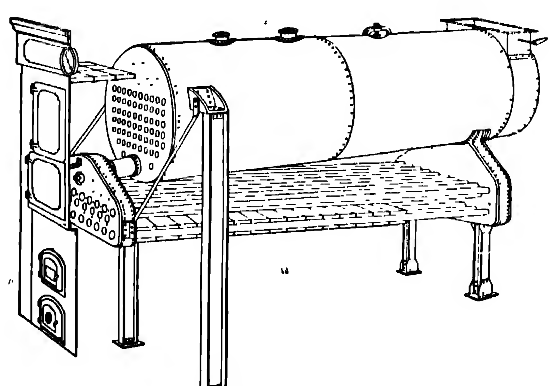

FIG. 49. Kroeschell Fire- and Water-tube Boiler.

heating surface to an 84-in. by 20-ft. unit containing 3480 sq. ft. A large number of Kroeschell combination boilers, in use in Chicago and its immediate vicinity, are giving excellent service.

72. Waste Heat Boilers. — The heat discharged by the waste gases of various industrial furnaces, such as open-hearth steel furnaces, brick and cement kilns, beehive coke ovens, metal reverberatory and refining furnaces, and the like, represent from 25 to 80 per cent of that of the fuel supplied. A considerable portion of this heat is reclaimed in the modern plant by so-called waste-heat boilers. Inasmuch as the temperature of waste gases available for this purpose varies from below 1000 deg. fahr., for long cement kilns, up to 2300 for melting furnaces, it is obvious that a boiler proportioned for direct fuel burning may be wholly unsuited for absorbing waste heat efficiently. With gases around 1000 deg. fahr., the heat transfer by radiation is almost negligible, while for temperatures around 2000 deg. fahr., the radiation is appreciable. Where the waste gases are discharged at a temperature above 2000 deg. fahr., the waste-heat boiler differs but little from that of a direct fired unit. However, the majority of waste-heat boilers in service are utilizing gases at temperatures ranging from 1100 to 1700 deg. fahr., and, since for this condition heat transfer is mainly by convection, the arrangement of heating surface and baffles must be modified to suit the new conditions. With low-temperature gases, to obtain a heat-transfer rate at all comparable with that found in ordinary boiler practice, the deficiency in temperature must be offset by an added velocity of the gases and an increased length of travel. Attention should also be called to the fact that the gases available for this class of work are almost invariably dirty; therefore, provision must be made for cleaning, by the installation of access doors, through which all parts of the setting may be reached. In many instances, settling chambers are provided for the dust before the gases reach the boiler. Furthermore, the operation of the boiler must in no way interfere with the operation of the primary furnace to which it is connected, and by-pass flues and dampers must be arranged so that the waste gases can be passed up the stack or to another waste-heat boiler. In order to obtain a high rate of heat transfer by increasing the velocity of the gases and the length of gas passages, the friction drop through the boiler becomes greatly in excess of what would be considered good practice in direct fired boilers, and mechanical draft is usually necessary. This is due to the fact that draft must be provided not only for the waste-heat boiler but for the requirements of the primary furnace. If the supply of waste heat is not continuous, as is frequently the case, it is customary to install auxiliary apparatus for direct firing.

72a. Reheat Boilers. — Boilers whose main function is to reheat the exhaust steam from high-pressure engines or turbines, and at the same time generate high-pressure superheated steam from water, are known as reheat boilers. Basically, they are not different from other boilers generating high-pressure superheated steam. See paragraph 405.

73. Heat Transmission Through Boiler Heating Surfaces. — All parts of the boiler shell, flues, or tubes which are covered by water and exposed to hot gases constitute the heating surface. Any surface having steam on one side and exposed to hot gases on the other is superheating surface. According to the A.S.M.E. Boiler Code, the side next the gases is to be used in measuring the extent of the heating surface. Thus measurements are made of the inside area of fire tubes and the outside area of water tubes. Each square foot of heating surface is capable of transmitting a certain amount of heat, depending upon the conductivity of the material, the character of surface, the temperature difference between the gases and the metal surface, the location and arrangement of the tubes, and the density and the velocity of the gases.

Figure 50 shows a section through a boiler-heating plate and serves to illustrate the accepted theory of heat transmission. The outer surface of the plate is covered with a thin layer of soot and a film of gas, and the inner surface is similarly protected by a layer of scale and a film of steam and water. It is, therefore, reasonable to assume that the dry surface of the plate is located somewhere within the film of gas, and the wet surface within the film of water and steam.

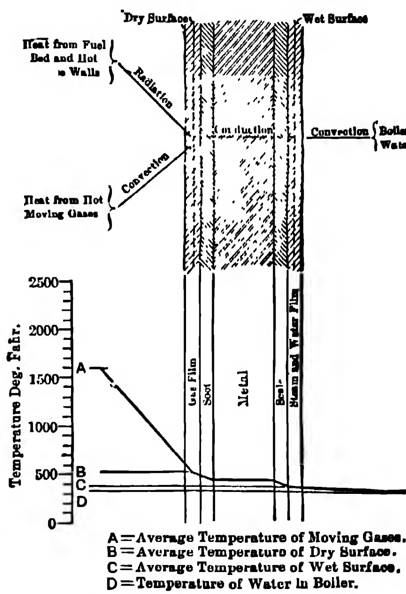

FIG. 50. Heat Transmission Through Boiler Plate.

The heat is imparted to the dry surface by: (1) **radiation** from the hot fuel bed and furnace walls, and by (2) **convection** from the moving furnace gases. The heat is transferred through the boiler plate and its coatings purely by **conduction.** The final transfer from the wet surface to the water is mainly by convection.

Radiation depends on the temperature, and, according to the law of Stēfan and Boltzmann, is approximately proportional to the difference between the fourth power of the absolute temperature of the fuel bed and

furnace walls and the temperature of the dry surface of the heating plate. According to this law, the heat transmitted by radiation increases rapidly with the increase in furnace temperature. Increasing the furnace temperature from 2000 to 3000 deg. fahr. will nearly quadruple the amount of heat imparted to the boiler by radiation. A drop in temperature from 2500 to 2400 deg. fahr. will reduce the amount about 12 per cent. The rate at which heat is radiated to the boiler heating surface is a function of the angle of exposure of the radiating surface, but is not influenced by the distance or shape of the boiler surface. With certain fuels, such as blast-furnace gas, wet wood, or bagasse, where the highest attainable temperature can be carried by the brickwork, it is best to absorb but little radiant heat, in order to maintain a high furnace temperature. With the higher-grade fuels it is necessary to absorb a considerable amount of the radiant heat in order to prevent too rapid destruction of the furnace refractories. In the modern boiler and setting, such as illustrated in Figs. 42 and 94, over 50 per cent of the total heat absorption at rating is from radiation.

For two " black bodies " with parallel faces exposed to each other, the heat transfer by radiation may be expressed by the equation

$$H = \frac{1600}{10^{12}} (T_1^4 - T_2^4) \tag{42}$$

H = B.t.u. per hr. per sq. ft.
1600 = radiation constant for black bodies
T_1, T_2 = temperature of the hot and cold bodies respectively, deg. fahr. abs.

In the actual furnace with water tubes, the rate of absorption will be from 0.2 to 0.3 of the theoretical, depending upon the nature of the surface. Consult " Radiant Heat," *Combustion*, March, 1926, p. 170.

In calculating the total heat transmission the resistance of the metal itself is so small that it may be neglected, and it may be logically assumed that the plate will take care of all the heat that reaches its dry surface.

In most boilers, where only a small portion of the heating surface is exposed to direct radiation from the incandescent fuel bed, and in waste-heat boilers, the greater part of the heat is transferred to the surface by convection.

The amount of heat imparted by convection from heated gases to cooler metal surfaces has been the subject of a great deal of investigation, both from the experimental and theoretical side. Numerous attempts have been made to correlate the experimental data with the theoretical deductions, but the results have been far from harmonious. This, however, has had little effect on the practical development of the boiler, and it is

quite possible that a more complete understanding of the phenomena will have no radical effect on the present design.

Experiments conducted by H. P. Jordan and the Babcock and Wilcox Co.[1] indicate that rate of heat transfer by convection in steam boilers varies approximately according to the law

$$U = K + BW/A \tag{43}$$

in which

U = coefficient of heat transfer, B.t.u. per hr. per sq. ft. per degree difference in temperature.
K = coefficient, determined experimentally.
B = a function of the dimensions of air passage and mean temperature difference of the gas and metal.
W = weight of gases flowing, lb. per hr.
A = average cross sectional area of the gas passages through the boiler, sq. ft.

The quantity W/A is called the **mass velocity,** and is approximately 1000 for the average boiler operating at rated capacity.

K is constant for a given design and ranges in value from 1 to 3.5. This constant (1) is greater for water tubes than for fire tubes of the same size, (2) is greater as the diameter of the tube decreases, and (3) is greater as the space between water tubes decreases. For the standard type of longitudinal-drum boiler, K is approximately 2.0 at 100 to 150 per cent rating.

B ranges in value from 0.0005 to 0.004 and (1) is greater for water tubes than fire tubes, (2) is greater as the diameter of the tube increases, and (3) is greater as the temperature difference between steam and gas increases. For the standard type of Babcock & Wilcox boiler, B is approximately 0.0014 at 100 to 150 per cent rating.

An examination of equation (43) shows that, for a given set of conditions and within certain limits, the rate of heat transfer varies directly with the weight of gases flowing per unit area of gas passage. This is not strictly true, since the rate of heat transfer varies as some power of the weight less than unity. But within narrow limits it is sufficiently accurate to consider the exponent as unity.

E. A. Fessenden[2] gives the following empirical formula for determining the drop in temperature of a gas flowing through a flue, which harmonizes

[1] Experiments on the Rate of Heat Transfer from a hot gas to a cooler surface published by the Babcock and Wilcox Co., New York, 1916.

[2] Trans. A.S.M.E., Vol. 38, 1916, p. 407.

the results of a large number of tests:

$$\text{loglog}\ (T_1/t_1) - \text{loglog}\ (T_2/t_1) = ML \tag{44}$$

wherein T_1 and T_2 are the mean absolute temperatures of gas at two points in the flue.

loglog = logarithm of the logarithm.
t_1 = mean abs. temp. of metallic wall, deg. fahr.
L = distance between points, ft.
M is a function of the rate of gas flow in the tube and the diameter, as given in Fig. 51.

The temperature varies directly with M; that is, it falls when a large flow of gas occurs, or with increased size of flues, which reduces the "hy-

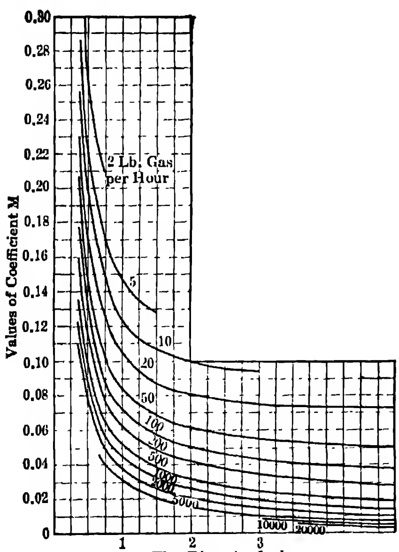

FIG. 51. Effect of Flow and Flue Diameter on Coefficient M.

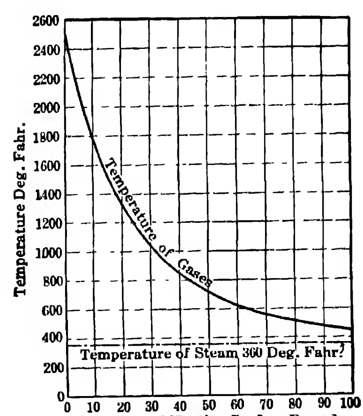

FIG. 52. Relation Between Temperature of Gases and Heating Surface Passed Over.

draulic depth." Low values of M mean lower efficiencies, although the total transfer may be increased.

The curve in Fig. 52 is based on Formula (44) for 2500 deg. fahr. initial, 450 deg. exit and 360 deg. steam temperatures.

Experiments by Prof. Nicholson[1] and the U. S. Bureau of Mines[2] show

[1] Proc. Inst. of Engrg. & Shipbuilders, 1910.
[2] Bul. 18, U. S. Bureau of Mines, 1912.

that by establishing a powerful scrubbing action between the gases and the boiler plate the protecting film of gas is torn off as rapidly as it is formed and new portions of the hot gases are brought into contact with the plate, thereby greatly increasing the rate of heat transmission. Similarly, the faster the circulation of the water, the greater will be the scrubbing action tending to remove the bubbles of steam from the wet surface, and the more rapid will be the transfer from the plate to the boiler water.

Professor Nicholson found that by filling up the flue of a Cornish boiler with an internal water vessel, leaving an annular space of only 1 in. around the latter, an evaporation eight times the ordinary rate was effected at a flow of gases 330 ft. per sec. (8 to 10 times the average flow). The fan for creating the draft consumed about 4 1/2 per cent of the total power.

The conclusion is that the heating surface may be reduced as much as 90 per cent for the same output under existing ratings, with a corresponding reduction in the size, cost, and space requirements, or, with a given heating surface of standard rating, the output may be enormously increased; also the increase in power necessary to create the draft is by no means comparable with the advantages gained.

The modern locomotive boiler is the nearest approach to these conditions in practice. Here a powerful draft forces the heated gases through small tubes at a very high velocity, and an enormous evaporation is effected with a comparatively small heating surface.

These principles have been applied to a limited extent to stationary boilers already installed, by making the gas passages smaller as compared to the length, and by forcing larger weight of gas through the boiler either by forced draft or by increasing the grate area.

The three distinct methods of heat transfer, radiation, convection, and conduction, do not exist separately in the modern steam boiler, but are operating at the same time. For this reason engineers find it convenient, for purposes of comparison, to consider the total heat transfer for the entire surface irrespective of the method of transmission. Assuming no losses in transmission, the actual heat exchange may be expressed

$$SUd = WCt = w\ (H_1 - q_2) \tag{45}$$

in which

S = sq. ft. of heating surface
U = *mean* coefficient of heat transfer, B.t.u. per sq. ft. per deg. difference in temperature per hr.
d = mean temperature difference between the heated gases and the metal surface, deg. fahr.
W = weight of gases flowing, lb. per hr.

C = average mean specific heat of the gases
t = mean temperature drop of the gases between furnace and breeching, deg. fahr.
w = weight of water evaporated, under actual conditions, lb. per hr.
H_1 = heat content of the steam, B.t.u. per hr.
q_2 = heat content of the feedwater, B.t.u. per hr.

The mean value of U varies within wide limits, as may be expected from the number of influencing factors. For surfaces exposed to direct radiation, U may be approximated from equations (42–44), and for surfaces receiving heat by convection only, equation (43) is ordinarily used.

For the average boiler operating at rated capacity, the mean value of U for the entire surface varies from 3 to 5. Modern high-set, coal-burning boilers, such as the one illustrated in Fig. 109, have values of U at maximum overload ranging from 6 to 12. In locomotive and marine boilers, U ranges from 8 to 20, and in waste-heat boilers from 2 to 15.

Example 19. — A boiler unit evaporates 32,000 lb. of water per hr. from a feedwater temperature of 200 deg. fahr. to steam at 200 lb. abs. pressure. Temperature of furnace, flue gas, and feedwater, 2550, 550, and 200 deg. fahr. respectively; heating surface 4000 sq. ft. Calculate the mean value of U.

Solution. — From steam tables, $H_1 = 1198$, $q_2 = 168$, temperature of the steam = 382.

$$d = (2550 + 550)/2 - 382 = 1168. \quad \text{(See paragraph 224.)}$$

Substituting these values in equation (45) and reducing

$$4000 \times U \times 1168 = 32{,}000\ (1198 - 168)$$

from which

$$U = 7.1$$

The maximum evaporation is limited only by the amount of coal which can be *burned*. For example, a *mean* evaporation as high as 23.3 lb. (22,600 B.t.u.) per sq. ft. of heating surface per hr. has been effected in locomotive work under intense forced draft, and 13.9 lb. (13,500 B.t.u.) per sq. ft. per hr. is not unusual in large central station boilers operating at peak loads. Such extreme high rates of evaporation, however, are invariably obtained at the expense of fuel economy. In the very latest central stations, the boiler and settings are proportioned to operate continuously at 200 to 300 per cent of standard rating with high overall efficiency and 450 per cent of rating for two or three hours, with only a small drop in efficiency, but such results are not obtainable in the ordinary furnace and setting.

Builders of return-tubular and vertical fire-tube boilers allow from 10 to 12 sq. ft. of heating surface per boiler horsepower (b.hp.); water-tube boilers are rated at 10 sq. ft. per b.hp., and Scotch marine boilers at 8 sq. ft. per b.hp.

Table 23 shows approximately the relation between b.hp. and heating surface for different rates of evaporation:

TABLE 23

RELATION BETWEEN EVAPORATION AND HEATING SURFACE

Evaporation from and at 212 Deg Fahr. per Sq. Ft. per Hr.										
2	2.5	3 0	3.5	4	5	6	7	8	9	10
Sq Ft. Heating Surface Required per B.Hp.										
17.3	13.8	11 5	9 8	8.6	6 8	5.8	4.9	4 3	3.8	3.5
B.t.u. per Hr. per Sq. Ft. Heating Surface (Thousands of B.t.u)										
1.9	2 4	2 9	3 4	3 9	4 8	5.8	6.8	7.7	8.7	9.7

The Transmission of Heat into Steam Boilers: U. S. Bureau of Mines, Bul. No. 18, 1912. *Radiant Heat:* Combustion, March, 1926, p 170.

74. Boiler Performance. — Tests of any kind, unless conducted in accordance with some accepted standard, are apt to be misleading and frequently are valueless for purposes of comparison. The accepted standard in the United States for testing power plant apparatus is that recommended by the Committee on Power Test Codes of the American Society of Mechanical Engineers, and published under the title "Rules for Conducting Performance Tests of Power Plant Apparatus." These rules, formulated by well-known specialists, give complete instructions regarding tests in general and a detailed analysis of the various items entering into the performance of boilers. The latest code is that approved March 11, 1924, and designated as "The Test Code for Stationary Steam Boilers, including Stokers, Superheaters, Economizers, and Air Preheaters." In the following paragraphs boiler performance will be analyzed in accordance with the old, or 1915, code, because practically all tests up to June, 1924, are based on these rules; but special attention is called to the particular items wherein the two codes differ. The changes in the new code are more in the matter of recording data than in the test methods themselves, though, as may be expected, the new code is the more comprehensive because of development in boiler-plant design. In the 1915 code a single form was used for all classes of boiler equipment and fuels; in the new code special forms are used for different combinations of boilers,

superheaters, economizers and air preheaters, and for solid, liquid and gaseous fuels. The subdivisions of the new, or 1923, code are as follows:

SOLID FUELS

Table 1*a*. Data and results, test of stationary steam-generating unit.
Table 1*b*. Heat balance of steam-generating unit. Short form.
Table 1*c*. Computations for test of stationary steam-generating unit.
Table 1*d*. Heat-balance computations, short form.
Table 2*b*. Heat balance of steam-generating unit comprising boiler and superheater, with or without integral economizers.
Table 2*d*. Heat balance computations for Table 2*b*.
Table 3*b*. Heat balance of steam-generating unit comprising boiler, superheater, and economizer.
Table 3*d*. Heat-balance computations for Table 3*b*.
Table 4*b*. Heat balance of steam-generating unit comprising boiler, superheater, economizer, and air heater.
Table 4*d*. Heat-balance computations for Table 4*b*.

LIQUID FUELS

Same tabular headings as for Solid Fuels.

GASEOUS FUELS

Same tabular headings as for Solid Fuels.

75. Units of Capacity. — According to the 1923 A.S.M.E. Boiler Code, the output of a boiler equipment may be expressed as:

(*a*) The weight of water fed to the boiler per hour at observed pressure and quality or temperature, and observed feedwater temperature.
(*b*) The equivalent evaporation per hour from and at 212 deg. fahr., and
(*c*) Boiler horsepower.

The weight of water fed to the boiler per hour under actual conditions is obtained directly from test measurements.

Because of the extreme range in practice in boiler pressure, quality, superheat, and feedwater temperature, the statement of pounds of water evaporated per hour gives no direct indication of the amount of heat absorbed, and for this reason a fixed condition of pressure, temperature, and quality is taken as a standard for comparison. The unit selected is the latent heat of vaporization of 1 lb. of steam at standard atmospheric pressure (14.7 lb. per sq. in. abs.). The latent heat under these conditions is the amount of heat absorbed by 1 lb. of water at a temperature

of 212 deg. fahr. in being converted to dry steam at the same temperature. The weight of water actually evaporated under observed conditions, expressed in terms of the weight which would be evaporated if the pressure were 14.7 lb. abs. and the feedwater temperature 212 deg. fahr., is called the **equivalent evaporation from and at 212 deg. fahr.** Thus it will be seen that the equivalent evaporation multiplied by 970.4 (Marks and Davis' value of the latent heat of vaporization at atmospheric pressure) gives the same total heat absorption as that under actual observed conditions. In order to facilitate transference from "actual" to "from and at 212," the **factor of evaporation** has been established. This factor, F, is the ratio of the heat absorbed by 1 lb. of feedwater under actual conditions to what it would absorb if the pressure were 14.7 lb. abs. and the feedwater temperature 212 deg. fahr., or,

$$F = \frac{H - q_2}{970.4} \tag{46}$$

in which

H = heat content of 1 lb. of steam at observed pressure and temperature or quality, B.t.u. per lb. above 32 deg. fahr.

q_2 = heat content of 1 lb. of feedwater at observed temperature, B.t.u. per lb. For most purposes $q_2 = t_2 - 32$, in which t_2 = temperature of the feedwater.

970.4 = latent heat of 1 lb. of steam at atmospheric pressure (Marks & Davis'). G. A. Goodenough's value is 971.7.

The values of H for saturated and superheated steam may be found in steam tables. For wet steam, H may be calculated from the relationship:

$$H = xr + q \tag{47}$$

in which

x = quality of the steam,

r = latent heat at observed pressure, B.t.u. per lb.,

q = heat of the liquid at observed pressure, B.t.u. per lb.

The boiler horsepower, b.hp., as originally defined (Centennial Rating) was based on a conventional engine water rate of 30 lb. of steam per i.hp.-hr. at 70-lb. gage pressure and feedwater at 100 deg. fahr. This is equivalent to 34.5 lb. of steam "from and at 212 deg. fahr." or 34.5 × 970.4 = 33,479 B.t.u. per hr.

The present, or A.S.M.E., standard b.hp. is the evaporation of 34.5 lb. of water from and at 212 deg. fahr. and, therefore, is the same in the

amount of heat absorbed as the Centennial b.hp. Since water rates vary from 5.5 lb. per hp.-hr., in the most economical grades of condensing engines using highly superheated steam, to 60 or 70 lb. per hp.-hr., in small non-condensing engines, it is apparent that the b.hp. has no connection whatever with the water rate of the engine. The power is developed in the engine, and the boiler itself does no work; therefore, the term b.hp. is purely arbitrary and misleading.

Manufacturers of stationary boilers ordinarily rate their boilers on the basis of 10 sq. ft. of heating surface per b.hp., and the power assigned is called the **builder's rating.** If used as an index of size only, "rated horsepower" is a satisfactory unit, but as the size of the boiler bears no relation to the horsepower of the engines it can furnish with steam, the term "horsepower" may well be omitted, and the extent of heating surface only be specified.

Example 20. — A boiler unit evaporates 30,000 lb. of water per hr. from feedwater at temperature 180 deg. fahr., to steam at 250 lb. abs. pressure and 600 deg. fahr. Required the factor of evaporation, equivalent evaporation, and b.hp. developed.

Solution. — From steam tables, $H = 1313.9$ and $q_1 = 180 - 32 = 148$. Substituting these values in equation (45) and solving for F.

$$F = (1313.9 - 148) \div 970.4 = 1.201.$$

The factor in this case signifies that it takes the same amount of heat to evaporate 1.201 lb. of water from a temperature of 212 deg. fahr. to steam at atmospheric pressure as it does to evaporate 1 lb. of water from a temperature of 180 deg. to superheated steam at 250 lb. pressure and 600 deg. temperature.

The equivalent evaporation per hr. is therefore 30,000 × 1.201 = 36,030 lb. and the b.hp. = 36,030 ÷ 34.5 = 1044. If the boiler is operated at a builder's rating of 10 sq. ft. of heating surface per b.hp., 10,440 sq. ft. of heating surface would be required. If operated at 200 per cent of rating, 5,220 sq. ft. would be required. In large central stations boiler units are frequently driven during peak loads at 350 to 450 per cent of rating.

Because of the variation in the value of the latent heat of vaporization at atmospheric pressure as given by different authorities, and the fact that computations must in any case be made in B.t.u., the Power Test Code Committee of the A.S.M.E. recommends that the capacity be given in heat units per hr. instead of b.hp., and that the round number 1000 be taken as the **unit of evaporation** instead of 970.4. For the data in Example 20, this would give the total heat output as 30,000 (1313.9 − 148) = 34,977,000 B.t.u. per hr. or 34,977,000 ÷ 1000 = 34,977 units of evaporation. It will be seen that the units of evaporation (using 1000

as a basis) are but 3 per cent lower than the equivalent from and at 212 deg. fahr., and offer the advantage of direct conversion into B.t.u. without calculation.

76. Units of Performance. — According to the final draft (March, 1923) of the revised code, the performance of a boiler (including firing equipment) should be expressed in terms of

SOLID FUELS

(1) Efficiency of boiler, superheater, furnace, grate and air heater: Ratio of heat units output to high calorific value of dry fuel or fuel as fired. (Omit " superheater " and " air heater " if the unit is not equipped with these appliances.)

(2) Efficiency, including economizer.

(3) Fuel (dry and as fired) per hr.; per sq. ft. of grate; per sq. ft. of retort and per burner per hr.

(4) Combustion space per lb. of fuel (dry and as fired) per hr.

(5) Actual and equivalent evaporation per lb. of fuel (dry and as fired) per hr.

(6) Equivalent evaporation per sq. ft. of heating surface per hr.

(7) Number of 1000 B.t.u. absorbed per sq. ft. of boiler heating surface per. hr.

(8) Boiler horsepower, average.

(9) Percentage rating.

The same items are used for liquid and gaseous fuels, except that the terms " grate " and " retort " are replaced by the term " burner."

Performance in terms of " combustible," as specified in the 1915 code, has been omitted from the revised code and is not mentioned in the code on definitions and values.

The more important of these items will be considered separately.

77. Boiler, Superheater, Furnace, Grate and Air-heater Efficiency. — A perfect boiler and furnace is one that transmits to the water in the boiler the total heat of the fuel and air. In order to effect this result, combustion must be complete, there must be no radiation or leakage losses, and the products of combustion must be discharged at approximately the initial temperature of the fuel. No commercial form of steam boiler can fulfill these conditions; hence the amount of heat absorbed by the boiler will always be less than the high calorific value of the fuel.

A general expression for overall efficiency of a steam-generating unit is

$$E = W\,(H_1 - q_2) \div H_f \tag{48}$$

in which

E = efficiency of the boiler equipment, consisting of boiler, superheater, furnace, grate, air heater, and economizer, or of as many of these appliances as are included in the equipment.

W = weight of feedwater evaporated into steam, at the observed pressure and quality, lb. per lb. of fuel as fired or dry.

H_1 = heat content of the steam, at observed pressure and quality, B.t.u. per lb. above 32 deg. fahr.

q_2 = heat content of the feedwater as fed into the boiler, B.t.u. per lb. above 32 deg. fahr. The efficiency is frequently expressed as "with or without economizer." The former is obtained by taking q_2 as the heat content of the water entering the boiler and the latter as that of the water entering the economizer. For all practical purposes, q_2 may be taken as $t_2 - 32$, in which t_2 = temperature of the feedwater, deg. fahr.

H_f = high calorific value of the fuel as fired or dry, depending upon the basis to which W is referred.

Test Code for Stationary Steam Generating Units (Preliminary Draft): Mech. Engrg., Sep. 1923, pp. 518-558.

The "efficiency based on combustible" was included in the 1915 Code but has been dropped in the revised Code. This efficiency is calculated from equation (48) by referring W to a "combustible as burned" basis and taking H_f as the calorific value of the "combustible."

The various items involved in the efficiency calculation of a steam-generating unit are best brought out by a concrete example. For simplicity, the unit is assumed to be without superheater, air heater, or economizer.

Example 21. — Calculate the capacity and economy of a steam-generating unit on the "as fired" and "combustible" basis, using the following data:

DATA AS OBSERVED

Boiler heating surface, sq. ft.	20,000
Builder's rating, hp.	2,000
Steam pressure, lb. per sq. in. gage	151.0
Barometric pressure, lb. per sq. in.	14.0
Steam pressure, lb. per sq. in. abs.	165.0
Temperature of feedwater, deg. fahr.	161.9
Temperature of flue gases, deg. fahr.	480.0
Temperature of boiler room, deg. fahr.	60.0
Quality of steam, per cent	98.0
Water actually evaporated, lb. per hr.	86,000
Coal as fired, lb. per hr.	10,000
Refuse removed from ashpit, lb. per hr.	1,600

COAL ANALYSIS, PER CENT OF COAL AS FIRED

Moisture	8
Ash	12

B.t.u per pound, 11,250.

Solution. — From steam tables, latent heat and heat of liquid of steam at 165 lb. abs. = 856.8 and 338.2 B.t.u. per lb., respectively. Therefore

Heat content of steam = $0.98 \times 856.8 + 338.2 = 1177.9$ B.t.u. above 32 deg. fahr.

Heat absorbed by 1 lb. of feedwater = $1177.9 - (161.9 - 32) = 1048$ B.t.u.

B.hp. = $86{,}000 \times 1048 \div (970.4 \times 34.5) = 2692$. Percentage of rated capacity developed, $100\,(2692 \div 2000) = 134.6$.

Factor of evaporation = $1048 \div 970.4 = 1.08$.

Water actually evaporated per lb. of coal as fired = $86{,}000 \div 10{,}000 = 8.6$ lb.

Equivalent evaporation per lb. of coal as fired = $8.6 \times 1.08 = 9.29$ lb.

Heat absorbed by the boiler per lb. of coal as fired = $9.29 \times 970.4 = 9015$ B.t.u.

Efficiency of boiler, furnace, and grate = $9015 \div 11{,}250 = 0.801$ or 80.1 per cent.

Refuse in ash referred to coal as fired = $1600 \div 10{,}000 = 0.16$ or 16.0 per cent.

Combustible burned on the grate referred to coal as fired = $100 - (8 + 16) = 76.0$ per cent.

Equivalent evaporation per lb. of combustible burned = $9.29 \div 0.76 = 12.22$ lb.

Heat absorbed per lb. of combustible burned = $12.22 \times 970.4 = 11{,}858$ B.t.u.

Combustible as fired = $100 - (8 + 12) = 80.00$ per cent.

Calorific value of the combustible as fired = $11{,}250 \div 0.80 = 14{,}062$ B.t.u.

Efficiency based on combustible = $(11{,}858 \div 14{,}062)100 = 84.3$ per cent.

Number of 1000 B.t.u. heat units absorbed per sq. ft. of boiler heating surface per hr. = $86{,}000 \times 1.08 \times 970.4 \div (20{,}000 \times 1000) = 4.5$.

Attempts have been made to separate the combined efficiency of boiler, furnace, and grate into two parts, viz., efficiency of the boiler alone and efficiency of the furnace and grate; but the results have been discordant and involve the use of factors which cannot be obtained with any degree of accuracy. Thus " true " boiler efficiency has been defined as the ratio of the heat absorbed to that available. The " heat absorbed " is taken as the difference between the heat generated in the furnace and that discharged into the flue, and the " available " heat is defined as the difference between the heat generated in the furnace and that discharged by the products of combustion at the temperature of the saturated steam.

If w_f, w_c = weight of the products of combustion in the furnace and passing through the uptake, respectively, lb. per hr.

T_f, T_c, T_s, T = absolute temperature of the furnace gases, flue gases, saturated steam and boiler room, respectively, deg. fahr.

c_f, c_c, c_s = mean specific heat of the products of combustion for temperature ranges t to t_f, t_c, t_s, respectively.

Then, neglecting radiation and minor losses, the "true" boiler efficiency equals

$$E_1 = \frac{w_f c_f T_f - w_c c_c T_c}{w_f c_f T_f - w_c c_s T_s} \tag{49}$$

Assuming no leakage, $w_f = w_c$; and neglecting the difference in the mean specific heats, $c_f = c_c = c_s$. With these assumptions, equation (49) reduces to

$$E_1 = \frac{T_f - T_c}{T_f - T_s} = \frac{t_f - t_c}{t_f - t_s} \tag{50}$$

The maximum theoretical efficiency, E_2, of the boiler or the efficiency of the *ideal* or perfect boiler, based on utilizing all the heat except the inherent losses, may be expressed as

$$E_2 = (H - I)/H \tag{51}$$

in which

H = calorific value of the coal as fired,
I = inherent losses as analyzed in paragraph 59.

The efficiency ratio, E_3, or the extent to which the theoretical possibilities are realized, may be taken as

$$E_3 = E/E_2 \tag{52}$$

in which

E = efficiency of the boiler, furnace, superheater, grate, air heater, and economizer, or of as many of these appliances as are included in the equipment.
E_2 = as in equation (51).

The furnace and grate efficiency, E_4, based on heat available, may be expressed

$$E_4 = [H - (I + F)] \div (H - F) \tag{53}$$

in which F = furnace losses, consisting of (*a*) loss due to unburned fuel dropping through the grate or withdrawn from the furnace, (*b*) loss due to the production of CO, (*c*) loss due to escape of unburned hydrocarbons, (*d*) loss due to the combination of carbon and moisture and production of hydrogen when fresh moist coal is thrown on a bed of white-hot coke, (*e*) radiation due to the furnace and (*f*) unaccounted for losses due to the furnace. (For an analysis of these losses, see paragraphs 49 to 60.)

Equation (53) does not furnish a method of finding the true efficiency, because it is impossible to determine loss (*d*) and impracticable to obtain loss (*c*) with the gas-testing appliances ordinarily available. It is also almost impossible to separate losses (*e*) and (*f*) attributed to the furnace from the "radiation and unaccounted for" losses attributed to the boiler alone.

TABLE 24

RELATION BETWEEN FUEL CONSUMPTION AND BOILER, FURNACE AND GRATE EFFICIENCY

(Pounds of Fuel Burned per B Hp -hr)

Calorific Value of Fuel, B.t.u. per Lb.	Boiler, Furnace and Grate Efficiency									
	40	45	50	55	60	65	70	75	80	85
7,500	11.17	9.91	8.94	8.12	7.45	6.87	6.37	5.95	5.58	5.25
8,000	10.45	9.30	8.37	7.60	6.97	6.13	5.98	5.58	5.22	4.92
8,500	9.84	8.75	7.87	7.12	6.56	6.05	5.62	5.25	4.97	4.63
9,000	9.30	8.25	7.45	6.76	6.20	5.72	5.31	4.96	4.65	4.36
9,500	8.80	7.83	7.05	6.40	5.87	5.41	5.02	4.69	4.40	4.14
10,000	8.37	7.44	6.70	6.09	5.58	5.15	4.79	4.46	4.18	3.94
10,500	7.98	7.09	6.39	5.80	5.86	4.90	4.56	4.26	3.99	3.76
11,000	7.60	6.79	6.09	5.52	5.06	4.67	4.34	4.05	3.80	3.59
11,500	7.28	6.49	5.83	5.29	4.85	4.47	4.16	3.88	3.64	3.45
12,000	6.97	6.22	5.58	5.06	4.65	4.28	3.99	3.72	3.48	3.28
12,500	6.69	5.97	5.35	4.86	4.46	4.11	3.82	3.57	3.34	3.14
13,000	6.44	5.74	5.15	4.68	4.29	3.96	3.68	3.43	3.22	3.02
13,500	6.20	5.52	4.96	4.51	4.18	3.81	3.54	3.31	3.10	2.91
14,000	5.98	5.33	4.79	4.35	3.99	3.68	3.42	3.19	2.99	2.81
14,500	5.77	5.15	4.62	4.20	3.84	3.54	3.30	3.08	2.88	2.72
15,000	5.58	4.96	4.47	4.06	3.72	3.43	3.19	2.98	2.79	2.64

In practice, the operating engineer is chiefly concerned with the combined efficiency of the boiler, superheater, economizer, air heater, furnace, and grate, as defined by the A.S.M.E. Boiler Code. This factor is readily determined with the ordinary instruments found in the average modern plant. In attempting to better the efficiency, it is necessary to separate the various losses as described in paragraphs 49 to 57, since this procedure enables the engineer to locate the source of loss, and, by comparing the actual and inherent losses, to show where improvement may be effected. Although efficiencies of 85 per cent or more have been

realized in several instances without the use of economizers or air heaters, such performances cannot be expected for continuous operation. In plants where there are no peak loads and the boiler may be operated under a constant set of conditions, a continuous efficiency of 83 per cent has been realized with bulk coal as fuel, and 85 per cent with fuel oil or powdered coal, but these figures are exceptional. In large central stations, having the usual peak loads in the morning and evening, and long banking periods, overall yearly efficiency is seldom greater than 78 per cent, though the boilers may be giving 80 to 86 per cent efficiency when operating at the most economical load. In large isolated stations with variable loads, an overall boiler and furnace efficiency on the yearly basis of 70 per cent is exceptional and a fair average is not far from 65 per cent. Small isolated stations, that show at times an efficiency as high as 75 per cent, seldom average 50 per cent for the year. In the small coal-burning house-heating plant, it is doubtful if the overall efficiency for the entire heating season exceeds 40 per cent. The preceding figures refer to boiler installations without economizers or air heaters. For influence of the latter on boiler,

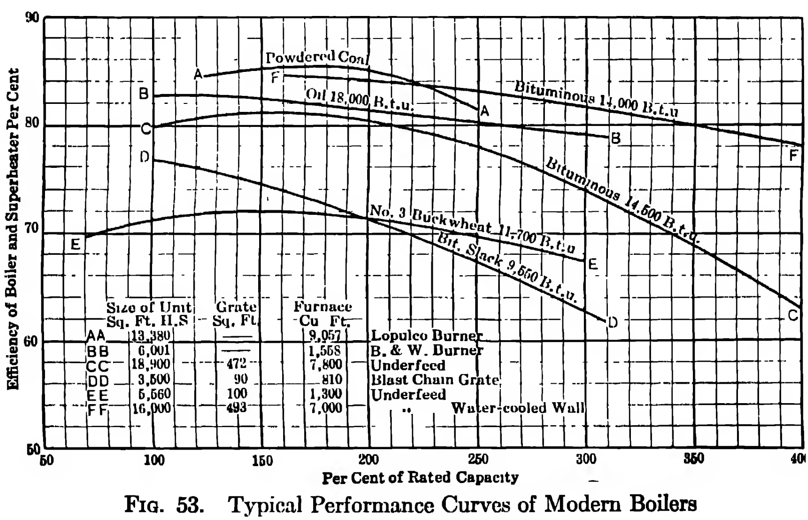

FIG. 53. Typical Performance Curves of Modern Boilers and Furnaces. No Economizers.

furnace, and grate efficiency, see paragraphs 261–3. In general, the overall efficiency is dependent primarily on the character of the fuel and the plant load factor. The greater the load factor, the smaller will be the standby losses (see paragraph 60), and the nearer will the overall efficiency approach test results. The usual discrepancy between efficiency as determined by special tests and average operation is due to the fact that the efficiency test is usually conducted under ideal con-

ditions. The boiler surfaces are cleaned, the rate of combustion carefully adjusted to maximum economy, and special attention given the firing, whereas, in most plants these refinements are seldom attempted. In our strictly modern boiler plants, refinement of design and a systematic supervision of operation have resulted in overall efficiencies far above anything hitherto thought possible.

TABLE 25

RELATION BETWEEN RATE OF EVAPORATION PER POUND OF FUEL AND BOILER, FURNACE AND GRATE EFFICIENCY

(Pounds of Water Evaporated per Hr. from and at 212 deg. fahr. per pound of Fuel)

Calorific Value of Fuel, B.t.u per Lb.	Boiler, Furnace and Grate Efficiency									
	40	45	50	55	60	65	70	75	80	85
7,500	3.09	3.48	3.86	4.25	4.64	5.02	5.41	5.80	6.18	6.57
8,000	3.30	3.71	4.12	4.55	4.95	5.36	5.77	6.18	6.60	7.01
8,500	3.51	3.94	4.38	4.81	5.26	5.70	6.14	6.57	7.01	7.45
9,000	3.71	4.18	4.64	5.10	5.56	6.01	6.50	6.96	7.42	7.90
9,500	3.92	4.41	4.90	5.39	5.88	6.17	6.86	7.35	7.85	8.33
10,000	4.12	4.64	5.16	5.66	6.19	6.70	7.21	7.74	8.25	8.76
10,500	4.31	4.86	5.40	5.94	6.48	7.01	7.55	8.10	8.64	9.17
11,000	4.52	5.09	5.65	6.22	6.79	7.35	7.91	8.48	9.05	9.61
11,500	4.74	5.31	5.91	6.50	7.10	7.69	8.28	8.86	9.45	10.0
12,000	4.94	5.55	6.16	6.78	7.40	8.01	8.64	9.25	9.86	10.5
12,500	5.14	5.78	6.42	7.06	7.70	8.35	9.00	9.64	10.3	11.0
13,000	5.35	6.01	6.69	7.35	8.01	8.69	9.35	10.0	10.7	11.4
13,500	5.56	6.25	6.95	7.65	8.34	9.03	9.72	10.4	11.1	11.8
14,000	5.75	6.48	7.20	7.91	8.64	9.35	10.1	10.8	11.6	12.2
14,500	5.96	6.70	7.45	8.20	8.95	9.70	10.5	11.2	12.0	12.7
15,000	6.18	6.95	7.72	8.50	9.26	10.1	11.8	11.6	12.4	13.1

The boiler, furnace, and grate efficiency is only one of the many factors entering into the economical operation of the boiler plant. Different fuels may give the same efficiency under actual operating conditions, but the ultimate economy in dollars and cents may vary considerably. The real criterion is the net cost of evaporation, taking into consideration first cost of equipment, the cost of handling the fuel, disposition of refuse, ability to handle peak loads, and depreciation of grate and setting. A popular, though somewhat empirical, method of comparing boiler performances is on the "fuel cost to evaporate 1000 lb. of steam from and at 212 deg." basis. The cost of fuel is taken as the total cost of fuel delivered to bunker or firing aisle plus the ash content, thus: if the cost of coal at the mine is \$1.95 per ton, freight \$1.90, handling \$0.50, ash content 16 per cent, the total cost is \$1.95 + \$1.90 + \$0.50 + \$0.16 = \$4.51. Each installation is a problem in itself, and all local influencing conditions

must be considered before maximum economy can be effected. In general, for plants equipped with coal and ash-handling machinery and adjacent to a railroad or to water transportation, the cheaper the fuel per pound of combustible, the lower will be the ultimate cost of evaporation.

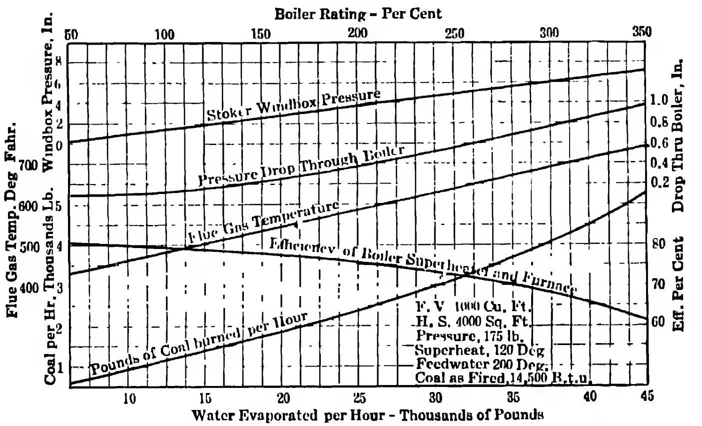

FIG. 54. Typical Performance Curves. Underfeed Stoker. No Economizer.

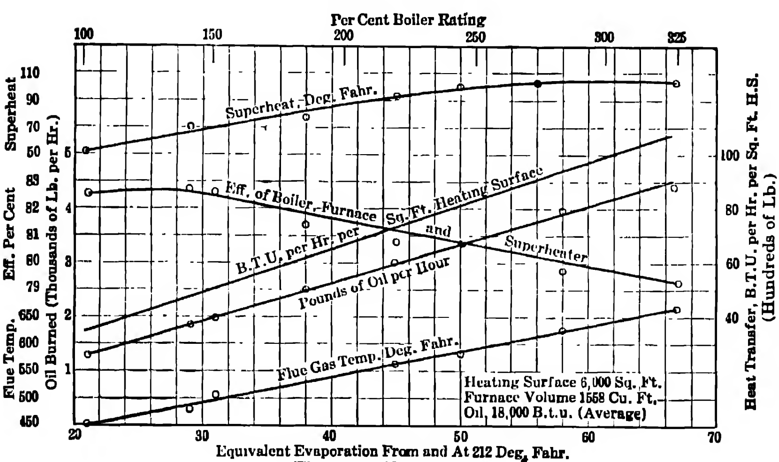

FIG. 55. Typical Performance Curves. Modern High-set Boiler. Mechanical Oil Burner. No Economizer.

78. Rate of Combustion. — According to the A.S.M.E. Boiler Code, the rate of combustion is expressed as (1) lb. of fuel (dry or as fired) per sq. ft. of grate surface, per sq. ft. of retort, per retort or per burner per hr. and (2) lb. of fuel (dry or as fired) per cu. ft. of furnace volume. Powdered, liquid, and gaseous fuels are burned in suspension, and since no grates

are employed the rate of combustion is expressed in terms of furnace volume or per burner only.

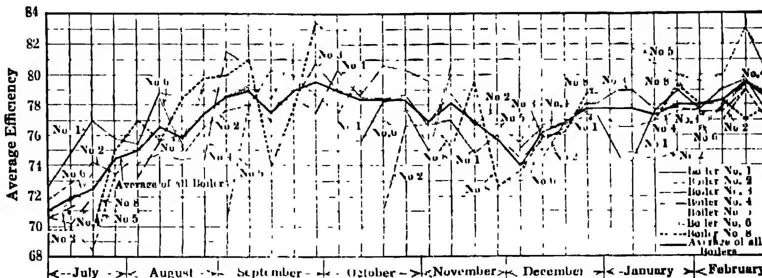

FIG. 56. Average Weekly Boiler Efficiencies. Colfax Station.

The capacity of a given boiler equipment is limited only by the amount of fuel which can be burned per unit of time. The rate at which solid fuel can be burned depends upon the extent and nature of the grate surface, character of the fuel and the draft. Efficiency of combustion is largely influenced by the size and proportional dimensions of the combustion chamber. In locomotive and marine practice, space limitations necessitate the use of small grates and combustion chambers, but in stationary plants there is a wide permissible range in size. In the former, the amount of fuel burned per sq. ft. of grate surface or per cu. ft. of furnace volume must be high, in order to obtain the desired capacity, but in the latter it may be high or low depending upon the design of the equipment.

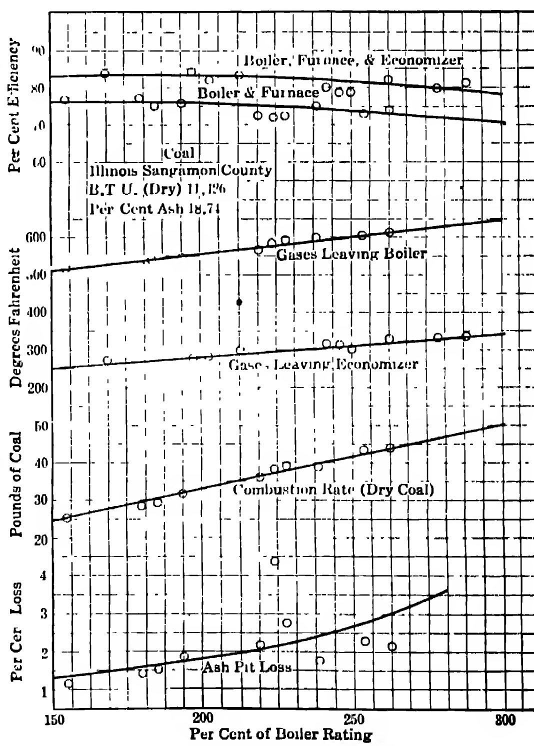

FIG. 57. Boiler Performance. Kansas City Power and Light Co. Forced-draft Chain Grate.

Grate Surface. Grate surface is defined by the A.S.M.E. Committee on Power Test Codes as the horizontal projected area of grates or stoker, including dump plates, ash crushers, etc. It is also stated as the total projected area of all surfaces supporting fuel, within the front wall of the furnace. In stationary practice there is a wide permissible range in proportioning grate surface, because a given rate of combustion may be effected with large grate surface and light draft, or with small grate surface and strong draft. For example, 9000 lb. of coal can be burned per hr. at a rate of 30 lb. per sq. ft. of grate surface per hr. on a 300 sq. ft. grate, and at a rate of 15 lb. per sq. ft. on a 600 sq. ft. grate. The draft necessary to force the air for combustion through the grate openings and fuel bed of the smaller grate, however, will have to be greater than that of the larger because of the increased depth of the fuel. By increasing the depth of the fuel and the draft pressure, any rate of combustion up to the maximum obtainable with that particular fuel can be maintained, provided the grate is correctly proportioned. There is a limit to the velocity at which air can be successfully forced through a given grate and kind of fuel. A certain time element is essential to the maintenance of combustion temperature and the proper mixing and contact of the air with the fuel and unconsumed gases. When the flow of air through the grate openings is so rapid that this time element is not provided, the fire is "blown out." This condition frequently arises when high rates of combustion are attempted with forced-draft stokers by raising the wind-box pressure until the velocity through the grate openings is excessive. The maximum rate of combustion is dependent upon the character of the fuel, stage of combustion, and provision for dissipating the air through the fuel bed. There is more danger of blowing out the fire in the ignition stage than after the carbon has reached incandescence.

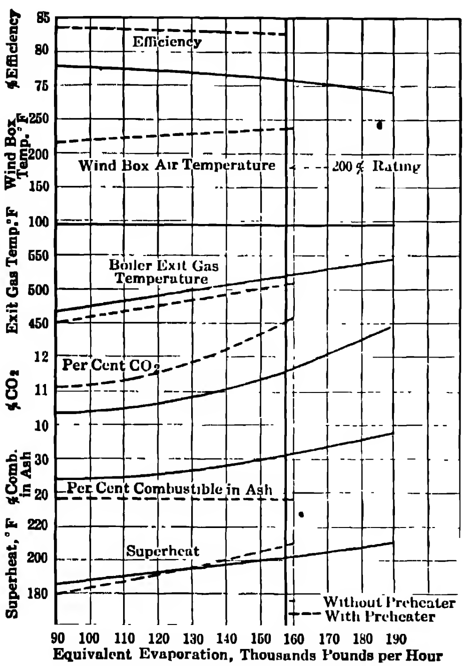

Fig. 58. Performance of Boiler No. 9, Colfax Station, with and without Preheated Air.

TABLE 26

AVERAGE RATES OF COMBUSTION

Lb. Fuel per Sq. Ft of Grate Surface per Hr.

(Bulk Fuel)

Kind of Fuel	Natural Draft		Forced Draft		
	Hand-fired	Chain Grate	Hand-fired	Chain Grate	Underfeed
Anthracite	15	Not suitable	20	45	Not suitable
Semi-anthracite.	16	30	25	45	40
Semi-bituminous .	18	35	35	40	40
Eastern bitum. .	20	35	30	45	45
Western bitum	30	35	35	50	50
Coke Breeze .	Not suitable	Not suitable	20	45	Not suitable
Lignite	25	40	35	50	40

TABLE 27

ECONOMICAL COMBUSTION RATES

(Worker and Peebles)

Fuel Analysis (Per Cent as Fired)		Eastern Coal	Pittsburgh Coal	Illinois Coal	Iowa Coal	Lignite
Fixed carbon .		73	57	48	33	34
Volatile .		17	30	30	27	35
Ash		6	7	12	25	10
Sulphur....		1	2	3	4	1
Moisture. . .		4	4	10	15	23
B.t.u. (dry)....		11,300	13,500	12,200	10,400	11,500
Combustion Rates: Lb. Dry Fuel per Sq. Ft. G. S. per Hr.						
Minimum for continuous operation	A	20–25	25–28	25–28	25–28	25–28
	B	15–18	18–20	18–20	18–20	18–20
	C	.	20–22	20–22	20–22	20–22
Recommended for continuous operation	A	30–38	32–40	30–38	28–35	30–35
	B	20–25	23–26	23–26	20–23	22–26
	C	.	23–26	23–26	22–25	25–30
Maximum for continuous operation	A	40–45	40–45	38–42	35–42	38–45
	B	25–28	30–35	30–32	25–27	26–32
	C		30–33	32–35	25–30	35–40
Recommended for 3 to 4 hour peaks	A	50–60	50–60	45–50	42–45	45–60
	B	30–35	35–40	32–35	27–30	32–35
	C		35–40	40–45	30–35	42–45
Maximum for 3 to 4 hour peaks	A	70	70	60	50	60
	B	40	42	40	30	35
	C	.	40	45	35	45

A Forced draft, underfeed.
B Natural draft, overfeed.
C Natural draft, chain grate.

Evidently the term "maximum rate of combustion per sq. ft. of grate surface per hr.," as determined by dividing the total maximum weight of fuel fired per hr. by the area of the grate, may be misleading, as for example in a forced-draft chain grate where there are three distinct stages of combustion. Here the weight of fuel burned is relatively small in the first stage, high in the middle and again low in the last. This illustrates the fact that air can be forced through certain sections of a grate at much higher velocities than through other portions of the same grate. The maximum economical rate of combustion of any fuel is largely influenced by the furnace and grate equipment. High rates of combustion usually result in high furnace temperatures with increased troubles from clinker formation, destruction of the furnace refractories,

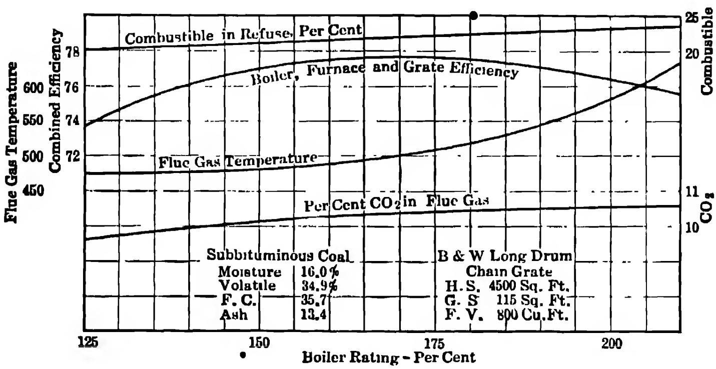

FIG. 59. Typical Performance Curve. Natural-draft Traveling-grate Stoker. Low-setting, Subbituminous Coal.

and burning out of tubes. For each fuel and grate there is a maximum rate of combustion beyond which the efficiency drops off rapidly, and the equipment should be designed to operate within this maximum. Where conditions permit, high rates of combustion should be avoided. In some of the latest power stations provided with water-cooled side walls, large combustion chambers, and clinker grinders, rates of combustion have been obtained at peak loads which a few years ago were thought impossible.

The ratio of grate area to heating surface is sometimes used as a guide in proportioning the grate, but the extent of grate surface depends upon so many factors that this method of procedure is of little value and is likely to lead to serious error. Thus, with anthracite and hand-fired grates, we find boilers operating successfully with ratios of grate surface to heating surface ranging from 1 to 30, to 1 to 60. With underfeed stokers burning bituminous coals, the range is from 1 to 35, to 1 to 60.

The curves in Fig. 60 give some idea of the relation between draft and rate of combustion for various fuels, and the values in Tables 26 and 27 offer a rough guide for estimating the average rates of combustion in general practice.

In locomotive and marine practice, rates of combustion as high as 225 lb. of coal per sq. ft. of grate surface per hr., have been attained, but such results cannot be considered seriously from an operating point of view. The results, however, show what can be done in the way of burning

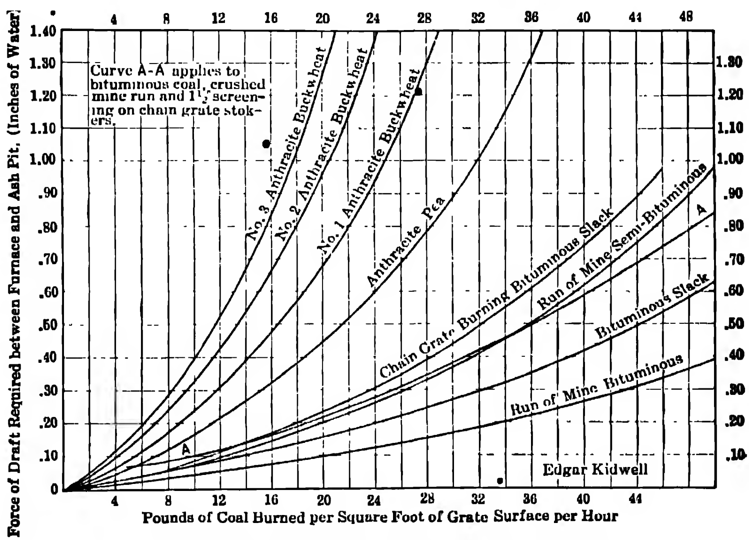

FIG. 60. Relation Between Draft and Rates of Combustion for Various Coals — Stationary or Traveling Grates.

solid fuel. In the latest large central stations, in which the boilers operate continuously at 200 to 250 per cent rating, the rate of combustion at these loads seldom exceeds 50–60 lb. of high grade bituminous coal per sq. ft. of projected grate surface per hr.

TABLE 28

MAXIMUM B.T.U. FIRED PER HR. PER CU. FT. OF FURNACE VOLUME AT EFFICIENCIES OF ABOUT 80 PER CENT WITHOUT ECONOMIZERS

Pulverized coal (ordinary type of furnace)	22,000
Chain-grate stokers (natural draft)	37,500
Chain-grate stokers (forced draft)	55,000
Underfeed stokers	64,000
Oil, steam atomizers	85,000
Scotch marine, hand-fired	144,000
Oil, mechanical burner	176,000
Pulverized coal (turbulent flow or well type furnace)	350,000

Furnace Volume. The maximum amount of fuel which can be burned efficiently per cu. ft. of furnace volume is a product of many variables. Among the more important factors may be mentioned the nature of the fuel, shape of furnace, type of fuel-burning equipment, and the means employed for mixing the air and volatile gases in the furnace. In view of the different conditions, general comparisons based on efficiencies and furnace volumes are of little value for purposes of design. Large furnace volumes mean increased first cost of settings and higher maintenance charges. Small combustion chambers are a necessity in marine and locomotive work, and high combustion rates must be maintained in order to realize the desired capacity. In stationary boilers there is no such restriction and there is a wide range in the size of furnace for a given rate of combustion. As much as 25 lb. of coal and 16 lb. of oil have been burned per hr. per cu. ft. of furnace volume in locomotive and marine boilers, but because of the reduced boiler efficiency such extreme rates of combustion are not to be considered except for emergencies. In the average coal-burning stationary plant, maximum combustion rates seldom exceed 2 lb. of coal and 4 lb. of oil per cu. ft. of furnace volume, but in modern central station practice as high as 4 lb. of coal and 8 lb. of oil have been burned per hr. per cu. ft. of furnace volume with good efficiency. See also paragraph 100. The data in Table 28 give the maximum B.t.u. fired per hr. per cu. ft. of furnace volume at efficiencies of about 80 per cent, without economizers, as recorded by Edwin B. Ricketts (Power, April 17, 1923, p. 613).

TABLE 29

COAL CONSUMPTION PER CU. FT. OF FURNACE VOLUME, LARGE CENTRAL STATION UNITS

Location	Boiler Heating Surface Sq. Ft.	Furnace Volume Cu. Ft.	Combined Eff. Boiler, etc., %		Coal, Lb. per Cu. Ft. of F. V.		Maximum Peak Load Per Cent
			Boiler Rating %		Boiler Rating %		
			100	350	100	350	
Seward	16,000	4800	78	67*	1 04	3.44	300
Hell Gate.....	18,900	7800	80	74	0 78	3.17	400
Delaware.....	15,000	5350	84	74	0 80	3 53	375
Hartford.....	13,920	6370	77	65	0 68	2 70	375
Colfax. ..	20,880	6420	81	74	1 07	3 54	300

* At 285 per cent rating

Tests of a Type W Stirling Boiler: P. W. Thompson, Trans. A.S.M.E., Vol. 44, 1922, p. 1005.

Boiler Room Performance and Practice at Colfax Station: C. W. E. Clarke, Trans. A.S.M.E., Vol. 44, 1922, p. 217.

Boiler Plant Efficiency, Victor J. Azbe, Trans. A.S.M.E., Vol. 43, 1921, p. 853.

Boiler and Furnace Economy: D. S. Jacobus, Trans. A.S.M.E., Vol. 43, 1921, p. 879.

79. Influence of Capacity on Efficiency. — Boilers are ordinarily rated on a commercial basis of 10 sq. ft. of heating surface per b.hp. This rating is absolutely arbitrary and implies nothing as to the limiting amount of water that this amount of heating surface will evaporate. It has long been known that the evaporative capacity of a well-designed boiler is limited only by the amount of fuel that can be burned on the grate. Thus, in locomotive practice, 1 b.hp. has been developed with 2 sq. ft. of heating surface, and in torpedo boat practice this figure has been reduced to 1.8 sq. ft. If there were no practical limitations to capacity, few, if any, boilers would be operated at the rated load, and the amount of heating surface for a given evaporation would be only a fraction of the present requirements. Briefly stated, the limitations are:

1. *Efficiency.* — As the capacity increases beyond a certain limit, the overall efficiency drops off, and a point is reached where further increase in capacity is obtained at a cost greater than that of additional heating surface.

2. *Grate Surface.* — All fuels have a maximum rate of combustion beyond which satisfactory results cannot be obtained. With this limit established, the only method of obtaining added capacity is through the addition of grate surface. Since the grate surface for a given boiler is limited by the impracticability of operating economically above a certain size, there is obviously a commercial limit to the maximum weight of fuel burned per unit of time.

3. *Draft.* — In order to effect a heavy rate of combustion, a great increase in draft is necessary. Apart from the power required to produce the draft, there is the possible loss of fuel carried away in the "cinders."

4. At heavy rates of driving, the furnace and stoker maintenance may become excessive.

5. *Feedwater.* — For continuous high boiler overloads, the feedwater must be practically free from scale-forming elements and matter which tend to cause foaming and priming.

6. *External Surfaces.* — Soot is such an excellent non-conductor of heat that provision must be made for its removal at frequent intervals, and, particularly so, if the boiler is expected to operate efficiently at heavy loads.

Tests show that if the furnace conditions are kept constant regardless of load, the efficiency of the boiler alone will decrease with increasing loads. But the furnace and grate efficiency increases with the capacity up to a certain point, beyond which it remains constant or gradually drops off. For a certain portion of the load, this increase in furnace efficiency may be at a greater rate than the decrease in boiler efficiency.

Consequently, the maximum combined efficiency may occur at a point either side of the rated capacity or remain constant over a considerable range of ratings.

In general, the combined efficiency of boiler, furnace, and grate increases with the capacity until a maximum is reached, from which point it drops off steadily with each increment of increase in load. This point of maximum efficiency varies with the type and size of boiler, kind of grate, design of furnace, character of fuel, and conditions of operation, and may range from 75 to 200 per cent or more of the rating. With stokers of the underfeed type, other things being equal, the highest efficiency is obtained from the greatest number of retorts, and the greatest effect on the overall efficiency is the rate of driving per retort. The curves in Figs. 54 to 60 are based upon authentic tests and give some idea of the effect of capacity on efficiency in specific cases. There are plants throughout the country in which boilers are developing, during periods of peak load, capacities of 500 per cent of the rating, and 603 per cent has been reached at the Hell Gate Station; but such loads cannot be maintained continuously (with the present type of equipment) with any degree of ultimate economy. It is a question if there are thirty plants throughout the country operating continuously day in and day out at 200 per cent rating. Widely varying loads are carried to-day in ordinary plant operation with overall efficiencies higher than those formerly secured from constant loads and under test conditions.

Some Comments on Boiler Capacity: L. R. Lee, Power, Mar. 15, 1922, p. 433.

80. Thickness of Fire. — For each boiler equipment, set of operating conditions, and grade of fuel, there is a depth of fuel bed which will give maximum efficiency; but, unfortunately, there are so many variables involved that general rules based on only two or three of the influencing factors are apt to be misleading. The composition and size of fuel, design of grate and stoker, type and size of boiler, method of firing, furnace construction, and general condition of the equipment exert such marked influence on the proper depth of fuel bed for a given rate of driving that actual tests of each installation are necessary before this item can be definitely established. For a given size and grade of fuel, a thick bed offers more resistance to the flow of air than a thin one; therefore, for a given draft pressure, the weight of air which can be forced through the fuel bed increases or decreases as the thickness is decreased or increased. Evidently there is a point beyond which increased depth will result in a deficiency of air, with accompanying reduction in capacity and efficiency. The reverse, however, is not true, since the greater the weight of air forced through the bed the greater will be the rate of combustion. Excess

air will be found above the unburned or devolatilized portion of the fuel bed, and immediately above holes in the fire, but all the air for complete combustion cannot be forced through the burning portion of bed from which the volatile matter is being distilled.

The following abstract from Technical Paper 80 U. S. Bureau of Mines, is of interest in connection with the *hand firing* of soft coals on stationary grates. "A thick fuel bed not only does not decrease the free oxygen in

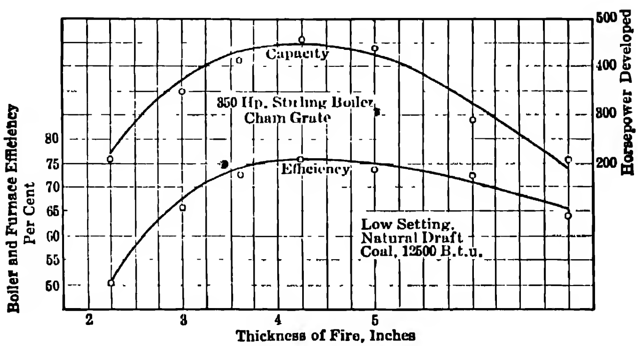

FIG. 61. Effect of Thickness of Fire on Capacity and Efficiency.

the flue gases, but it may actually increase it, thus: Assume that in a hand-fired furnace with a fuel bed 5 in. thick, the quantity of air admitted through the proper opening in the fire doors is sufficient to burn completely the combustible gases rising from the fuel bed. Now, if the thickness of the fuel bed is increased to 10 in., its resistance is nearly doubled; the draft over the fuel bed is increased somewhat, but is not doubled. The quantity of combustible gases rising from the surface of the fire depends directly on the quantity of air flowing through the fuel bed. Therefore, when the resistance of the fuel bed is nearly doubled by doubling the thickness of the fuel bed, less air (but more than one-half) flows through the fire and less combustible gas rises from its surface. At the same time the openings admitting air over the fuel bed remain constant, so that the higher furnace causes more air to flow over the fire. Thus, when the fuel bed is 10 in. thick, less combustible gases are burned with larger air supply over the fire than when the fire is only 5 in. thick, provided, of course, that in both cases the fire is perfectly level and is free from holes.

The accumulation of clinker has the same effect as thickening the fuel bed. The clinker increases the resistance to the flow of air through the fuel bed, so that the latter generates a smaller quantity of gas. The increased draft in the furnace draws in more air through the openings in the fire door, so that more is used to burn 1 lb. of coal when the grate is

clinkered than when the fire is clean. This fact is known to every boiler-room operator.

Many firemen do not like thin fuel beds because they cannot run the fires with long intervals between firings. However, this feature is rather in favor of a thin fuel bed than against it. If the fireman must give the fires frequent attention he is more likely to keep them level and free from holes. A thin and level fuel bed is the most important requisite in burning coal efficiently.

A thick fuel bed is a common cause of excessive clinkering, particularly in the case of a coal whose ash melts at relatively low temperature. Clinker

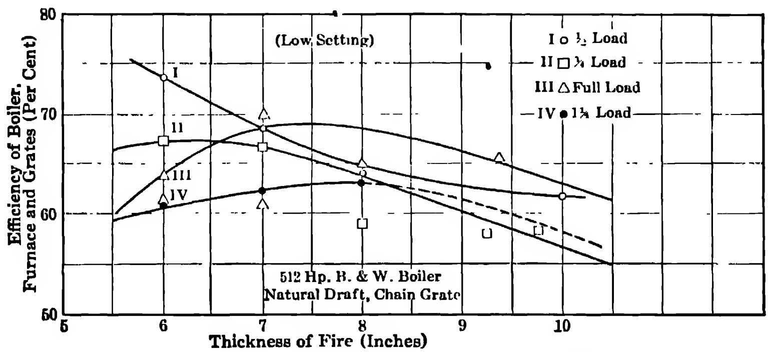

FIG. 62. Influence of Thickness of Fire on Efficiency.

forms in thick fuel beds because the reduced air supply through the grates permits the ash to become heated and because the heating is partly done in a reducing atmosphere of CO.

Under the usual natural-draft operating conditions in stationary plants equipped with hand-fired stationary grates and burning soft coal, there is no reason why fires should be carried thicker than 8 in. and with some coal even an 8-in. fire is too thick. If the coal is coarse and contains only a small portion of fine coal, the thickness of the fuel bed may be near 8 in.; but if the coal is mostly small pieces and slack, better results are obtained with the thickness of the fuel bed near 4 in."

With chain-grate stokers, the depth of fuel bed in average practice ranges from 4 to 12 in. depending upon the draft, nature of the fuel, and speed of the grate. With underfeed stokers the depth may range from 10 in. to 2 ft. Because of the increased agitation of the fuel bed at heavy ratings, the resistance through the fuel of an underfeed stoker may be less at maximum load than at somewhat lower loads. See Fig. 239. As previously stated, the most economical thickness of fire can be determined only by actual test of each installation. Some idea of the influence of

thickness of fire on the efficiency and capacity in specific cases may be gained from the curves in Figs. 61 and 62.

81. Pressure Drop through Boilers. — The resistance encountered by the gaseous products of combustion in passing through the boiler results in a pressure drop or " draft loss," which varies greatly with the type and size of boiler, arrangement of gas baffles, number of gas passes, design of superheater, amount of air used per lb. of fuel, and the rate at which the boiler is operating. For a given equipment, the pressure drop from furnace to uptake varies approximately with the square of the velocity of flow.

The vertical passes in any boiler act as chimneys and are capable of furnishing a draft pressure in much the same manner as the chimney proper. The greater the length of the vertical passes the greater will be the " chimney action." The pressure difference due to the chimney action may decrease or increase the draft of the stack, depending upon the

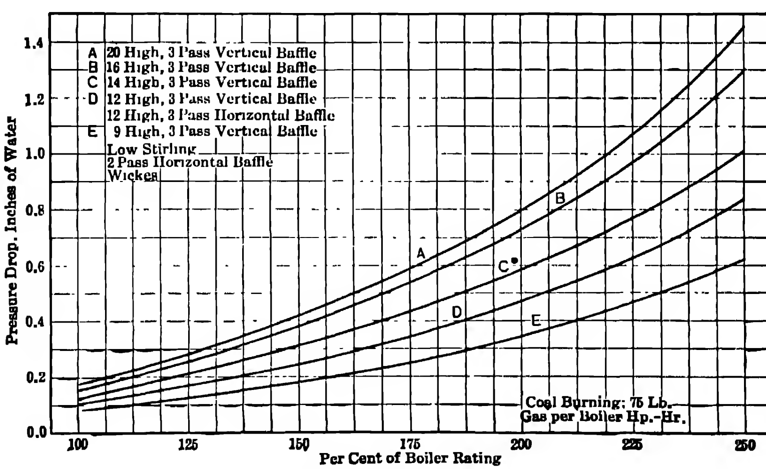

FIG. 63. Pressure Drop through Boilers.

direction of flow of the gases. If the flow is upward, the vertical pass acts as an additional height of stack; if downward, it tends to retard the flow. Thus, in the Wickes boiler, Fig. 43, the vertical path of the gases through the boiler itself causes considerable chimney action. At low rating the pressure at *C* may be atmospheric or even slightly above, although the draft in the combustion chamber *B* may be 0.10 in. of water below that of the atmosphere. This means that the boiler itself furnishes sufficient chimney action to operate the boiler at this load. Similarly, the draft at *D* may be higher than at *C* because of the negative chimney action and resistance combined. The difference in temperature of the

gases, due to the cooling action of the heating surface, must of course be considered in calculating the chimney action. In practically all boilers, the chimney action of the vertical passes influences the pressure drop throughout the setting, and the effect is more marked when the rate of flow is low.

TABLE 30

AVERAGE PRESSURE DROP THROUGH BOILERS

Boilers Operating at Rating

(Frank R. Chambers)

Type	A	Type	A
Atlas (Horizontal Pass)	75	Heine (2 Pass Horizontal)	65
Atlas (Vertical Pass)	60	Keeler (Vertical Pass)	51
B. & W. (Vertical Pass)	50	Keeler (Horizontal Pass)	55
B. & W. (Sewall Pass)	65	Oil City (Vertical Pass)	60
B. & W. (Horizontal Pass)	60	Page	55
Cahall (Vertical)	65	Return Tubular	45
Edge Moor (4 Pass Vertical)	64	Scotch Marine	65
Edge Moor (3 Pass Vertical)	55	Stirling (5 Pass)	81
Edge Moor (Horizontal Pass)	60	Stirling (4 Pass)	75
Erie City (Vertical)	60	Stirling (3 Pass)	65
Erie City (Horizontal)	60	Wickes (Vertical)	58

A = Pressure drop through boiler, per cent of total draft at stack side of damper. This factor applies only to hand-fired furnaces burning about 25 lb. Illinois coal per sq. ft. of grate surface per hr.

Because of the great variation in the size and design of boilers, the variety of baffle arrangement, and the wide range in operating conditions, it is impossible to establish rules for draft losses which can be of general application, and it is advisable to obtain specific data from the manufacturers. The values in Table 30 are based upon the investigations of Frank Chambers, Deputy Smoke Inspector of the Department of Health, Chicago, Illinois, and give some idea of the draft losses through hand-fired boilers operating at rated capacities when burning bituminous coal at approximately 25 lb. per sq. ft. of grate surface per hr.

The curves in Fig. 63 show how the draft losses vary with different types of boilers at various ratings when burning bituminous coal, heating value 12,500 B.t.u., with 100 per cent air excess. The curves are applicable only to the specific cases analyzed, but may be used as rough approximations for preliminary calculations.

The draft loss with forced-draft chain grate and underfeed stokers is roughly 20 per cent less than that given in the curves of Fig. 63, and with oil fuel about 25 per cent less. With blast-furnace gas, the draft loss is about 15 per cent greater than that given in the curves.

82. Flue-gas Temperatures. — The sensible heat carried away by the flue gas is usually the greatest loss in the generation of steam. The greater the extent of heat-absorbing surfaces for a given weight of gas, the lower will be the loss. In the ordinary boiler without preheating or economizer surface, the minimum theoretical temperature of the flue gas is that corresponding to the temperature of the steam. With preheaters or economizers the minimum theoretical temperature is that corresponding to the lowest temperature of the heat-absorbing fluid. While it is a comparatively simple matter to add sufficient heat-absorbing surface to reduce the flue-gas temperature to nearly the theoretical limit, such a procedure is not warranted by the present cost of fuel. Fixed charges and operating and maintenance costs more than offset the gain. The heating

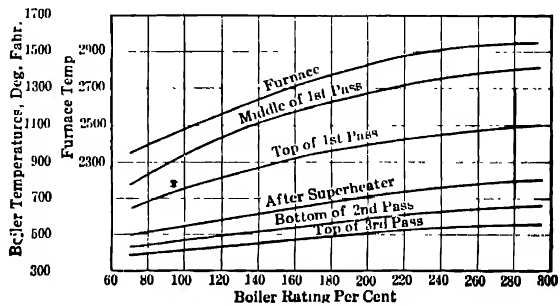

FIG. 64. Influence of Rate of Driving on Boiler and Furnace Temperatures.

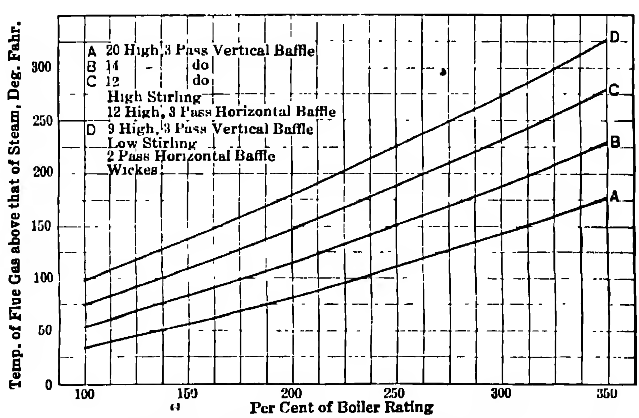

FIG. 65. Influence of Rate of Driving Flue Gas Temperatures.

surface in the modern boiler without preheating element or economizer is usually proportioned and baffled so that the exit temperature of the flue gas at boiler rating is from 25 to 100 deg. fahr., above that of the saturated steam.

Flue-gas temperatures are functions of the composition of the fuel, air excess, rate of driving, arrangement of baffles, extent of heating sur-

face exposed to direct radiation, cleanliness of the heating surfaces, and design of boiler, superheater, and furnace. Some idea of the temperature ranges in various parts of the setting of a modern stoker-fired furnace may be gained from the curves in Fig. 64. The curves in Fig. 65 are only approximate and should not be used for purpose of design. Specific data for any set of operating conditions may be had from boiler manufacturers. For a given boiler and furnace equipment and rating, flue-gas temperatures are generally lower with oil and gaseous fuels than with solid fuels because of the smaller air excess, and for the same reason mechanical stokers give lower temperatures than hand-firing. Flue-gas temperatures for a number of specific cases are given in Figs. 54, 55 and 58.

83. Economical Loads. — The most economical rating at which a boiler plant can be run depends primarily upon the load to be carried by that individual plant and the nature of such load. The most economical load from a commercial standpoint is not necessarily the most efficient load thermally, since first cost, cost of upkeep, labor, cost of fuel, capacity, and the like must all be considered along with the thermal efficiency. The controlling factor in the cost of the plant, that is, the number of boiler units that must be installed, regardless of the nature of the load, is the capacity to carry the maximum peak loads. While each individual set of plant operating conditions must be considered by itself, the following statements give some idea of general practice:

For a constant 24-hr. load, the operating capacity, to give the highest overall plant economy, is from 25 to 75 per cent above that incident to maximum thermal efficiency.

For the more or less constant 10- or 12-hr. a day load, where the boilers are placed on bank at night, the point of maximum economy will be somewhat higher, probably from 50 to 125 per cent above that incident to maximum thermal efficiency.

The third class of load is the variable 24-hr. load found in central station work.

Modern methods of handling loads of this description, to give the best operating results under different conditions of installation, are as follows:

1. The load on the plant at any time is carried by the minimum number of boilers that will supply the power necessary, operating these boilers at capacities of 150 to 250 per cent or more of their normal rating. Such boilers as are in service are operated continuously at these capacities, the variation in load being cared for by varying the number of boilers on the line, starting up boilers from a banked condition during peak load periods and banking them after such periods. This is, perhaps, at present the most general method of central station operation.

2. The variation in the load on the plant is handled by varying the capacities at which a given number of boilers are run. At low plant loads, the boilers are operated somewhat below their normal rating, and during peak loads, at their maximum capacity. The ability of the modern boiler to operate over wide ranges of capacities without appreciable loss in efficiency has made such a method practicable.

3. The third method of handling the modern central station load is, perhaps, only practicable in large stations or groups of inter-connected stations. Under this method, the plant is divided into two parts. What may be considered the constant load of the system is carried by one portion of the plant, operating at its point of maximum economy. Due to the possibility of very high overall efficiencies at high boiler capacities where the load is constant, where the grate and combustion chamber are designed for a point of maximum economy at such capacities, and where there are installed economizers and such apparatus as will tend to increase the efficiency, the capacity at which this portion of the plant is to-day operated will be considerably above the point of highest economy for the steady 24-hr. load for boilers without economizers.

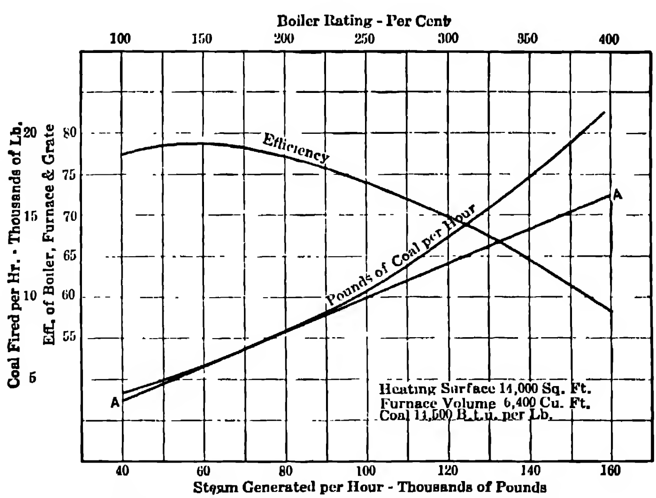

FIG. 66. Performance of Boiler and Furnace.

The variable portion of the load on a plant so operated is carried by the second division of the plant, under either of the methods of operation just given.

The problem involved in deciding whether to force boilers over the peak or bank additional boilers may be analyzed as follows: Suppose the curves in Fig. 66 are representative of the performance of the boilers in a

large central station and it is desired to establish a general relationship between economical forcing and banking hours. An inspection of the curves in Fig. 66 shows that the maximum thermal efficiency is at a steaming rate of 40,000 lb. per hr., corresponding to 100 per cent rating. Suppose, however, that a steaming rate of 70,000 lb. per hr., corresponding to 175 per cent rating, has been demonstrated to be the most economical and practical for operation through the day. (This point can be determined only by actual operation, taking into consideration first cost, attendance and maintenance as well as thermal efficiency.) By forcing the boilers over the peak to a higher rate, the plant may be run with fewer boilers banked while operating on the day and night loads, thus saving

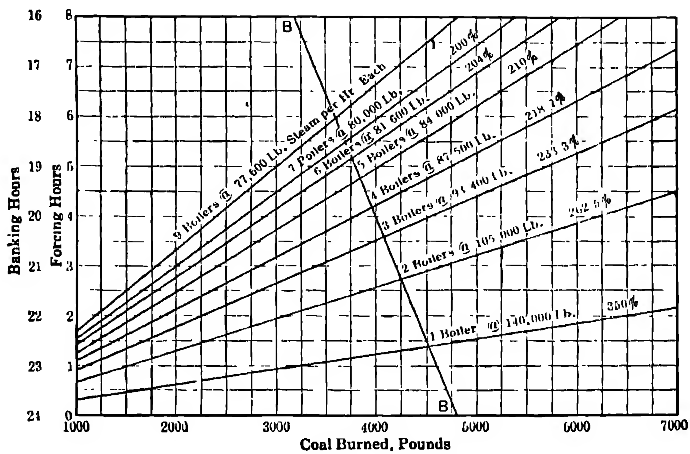

FIG. 67. Curve Showing Extra Coal Burned for Forcing and Banking.

banking losses, but this gain is offset to a certain extent by the extra amount of fuel required to maintain the higher ratings. Evidently, there is a point at which the loss due to the extra fuel burned by forcing is equal to that gained in reducing the banking period. Referring to Fig. 66, the line AA represents the amount of fuel burned at various steaming rates on the supposition that the rate of fuel consumption at 175 per cent boiler rating is maintained throughout the entire range of operation. The difference between the actual fuel consumption line and the line AA represents the extra amount of fuel burned at any particular load. Thus at 350 per cent rating the coal burned is 17,500 lb. per hr. as against 14,300 lb. per hr. at 175 per cent rating, or 3200 lb. per hr. in excess of that required to operate two boilers at 175 per cent rating.

The diagonal lines in Fig. 67 represent the loss in pounds of coal caused by forcing different combinations of boilers above 175 per cent rating in

order to carry the same load as this combination plus one more boiler, all boilers in the latter combination operating at 175 per cent rating. Thus, one boiler at 350 per cent rating generates the same quantity of steam as two boilers at 175 per cent rating and burns 3200 lb. per hr. more coal than the latter combination; two boilers at 262.5 per cent rating have the same capacity as three boilers at 175 per cent and burn 1550 lb. per hr. more coal than the latter combination; three boilers at 233.3 per cent have the same steaming capacity as four boilers at 175 per cent and burn 1137 lb. per hr. more coal than the latter combination; and so on. Line BB, Fig. 67, represents the weight of banking coal burned by one boiler for the period indicated and is based on the assumption that the coal burned in banking is at the rate of 200 lb. per hr. The intersection of the "forcing" line with the "banking" line is the point at which the extra coal for forcing is equal to that burned in banking. The curve in Fig. 68 is obtained by plotting the hours, as found from the point of intersection, Fig. 67, against the steaming rate or per cent boiler rating. This curve shows the limit of forcing beyond which the losses are greater than the gains. For example, the boilers should not be operated above 350 per cent rating for more than 1.4 hr., or above 300 per cent rating for 2 hr., and so on. No provision has been made for furnace maintenance, which at very high ratings may be excessive. This may be included by allowing an additional weight of fuel for forcing to compensate for the extra cost of maintenance. The curve in Fig. 68 is not general and is applicable only to the specific case under consideration; the method, however, is general.

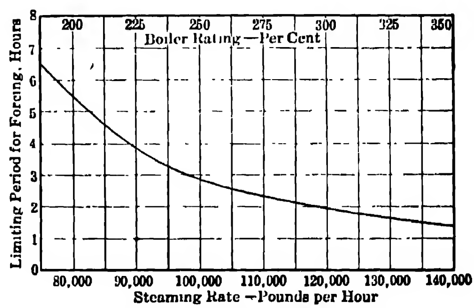

FIG. 68. Curves Showing the Limit of Economical Forcing.

Present Day Boiler Room Operation: I. E. Moultrop and R. E. Dillon, Power, Mar. 7, 1922, p. 384.

Refinements of Practice in Modern Power Plants: I. L. Kentish-Rankin. Power Plant Engineering, Oct. 15, 1921, p. 988.

Efficient Operation of the Boiler Plant: J. D. Morgan, Power, June 15, 1920, p. 957.

Development of Power from the Standpoint of the Boiler Room: C. F. Hirshfeld, Power, Aug. 20, 1918, p. 284.

84. Selection of Type. — Boilers constructed by builders of good repute are usually designed for safety, durability, and capacity, and rigid

specifications and inspection of material and workmanship on the part of the purchaser are ordinarily not necessary, as the makers' reputations are sufficient guarantee of their worth. Marked departure from standard designs must necessarily be specified, and must comply with the state and community boiler laws and insurance requirements; but in most cases instructions are limited to the working pressure, extent of heating and grate surface, the character of the furnace, and arrangement of setting. Numerous tests on various types of boilers show practically the same efficiency provided the furnaces and boilers are properly designed, so that the relative merits may be considered with reference to (1) durability; (2) accessibility for repairs; (3) facility for cleaning and inspection; (4) space requirements; (5) adaptability to the type of furnace and stoker desired; (6) overload capacity; and (7) cost of boiler and setting. For rated capacities above 200 hp. and pressures above 150 lb. per sq. in. or more, the water-tube or some form of internally fired boiler in which the shell plates are not exposed to the high temperature of the furnace is considered safer than the horizontal tubular boiler, because the shell plates and the seams of the latter must be of considerable thickness in the larger units, and being exposed to the hottest part of the fire are likely to give trouble, especially if the water contains scale or sediment-forming elements. In the modern central station, steam pressures of 275 to 350 lb. per sq. in. are standard practice. In a few recent installations, a pressure of 550 lb. has been specified, and at least two plants have placed orders for boilers to operate at 1200 lb. gage. (See paragraphs 183 and 214 for a discussion of high pressures.) Return-tubular and stationary locomotive boilers are seldom made in sizes over 250 hp. and hence are not to be considered for large units. For sizes under 200 hp. (78-in. by 20-ft.), the return-tubular boiler is most commonly installed, unless high pressure and low head-room is essential, in which case the internally fired Scotch-marine boiler or a cross-drum type of water-tube boiler, such as the Burton, is used. The water-tube boiler is usually employed in large central stations for high-pressure units of 200 to 3000 hp.

The particular type of water-tube boiler is to some extent a matter of personal taste on the part of the engineer, but due consideration should be given to the special requirements as listed above. For small powers and for intermittent operation, small vertical or horizontal fire-box boilers have the advantage of low first cost. The small air leakage and radiation losses give internally-fired boilers an advantage over the brick-set externally-fired fire-tube or water-tube types, but this is partly offset by the greater extent of regenerative surface in the setting of the latter. In several recent installations, the brick settings are completely encased in steel, and a layer of high-grade insulating material is placed between the

brickwork and the casing. This reduces the leakage and radiation losses to a minimum, and the setting remains effective over a long period of time. Internally-fired boilers are more expensive than the externally fired, though the extra cost of setting and foundation in the latter may bring the total cost of the entire equipment to practically the same figure. Internally-fired boilers above 300 hp. rated capacity are not much in evidence in stationary plants. The design and installation of the boilers and furnaces should be left at the outset to a capable engineer.

Makers usually request the following information from intending purchasers:

1. The kind of fuel to be burned.
2. The type of furnace or stoker.
3. Head room.
4. Steam pressure and superheat desired.
5. The quantity of steam demanded.
6. The nature and intensity of draft.
7. Quality of feedwater.
8. Class of labor procurable.
9. Characteristic load of plant.

85. Selection of Size. — The most economical size of individual boiler units for any plant is dependent primarily upon the maximum steam requirements and character of the load. The load curve for manufacturing plants may be predetermined with a fair degree of accuracy, since the power and steam demands for various purposes may be readily segregated and analyzed; but with public utility concerns and certain classes of isolated stations the problem is largely a matter of experience and judgment. The load curve should include not only the average yearly load, but also the maximum daily load which is likely to occur, the minimum daily load, temporary peak loads, and probable future increase. In most cases the general characteristics of the load curves are based upon those of similar plants having comparable conditions of operation, and the magnitude of the load is calculated from the power and steam requirements of the particular plant under analysis. As water rates of prime movers and various auxiliaries may be obtained from the manufacturers, the steam consumption at various loads may be readily calculated from the assumed load characteristics.

With the steam requirements known, the next step is the determination of the number and size of boiler units to be installed. In the first place, all boilers in a plant should be of the same size and type if possible, to insure uniformity of equipment and operating methods. Thermal efficiencies are usually higher, labor costs lower, and first cost of the entire boiler equipment less for a few large boiler units than for a number of

small units of the same total capacity; therefore, the units should be of the largest possible size compatible with the size of the plant and operating conditions, and the total steam requirements should be divided among such a number as will give proper flexibility of load and insurance against interruption of service. As all boilers have to be shut down at times to allow for cleaning and repairs, standby units or "spares" are usually necessary to carry the load when the others are out of service.

In the average plant of, say, 500 to 2500 hp. with fairly good fuel and feedwater, three boilers are ordinarily installed, two to carry the load and one to stand in reserve; but where frequent cleaning is necessary and continuity of operation is essential, two spares may prove to be the better investment. In large central stations, the boiler plant is usually laid out on the unit or panel system, each panel serving one prime mover. In the older designs, each panel has 6 to 10 boilers, including spares, even though the sections are cross connected; but in the very latest designs there are but two or three boilers per turbine unit and there are no spares. Some attention has been given to the one-turbine one-boiler idea, with a view toward simplifying plant and piping design and reducing boiler-room labor, but as yet no such installation has been made in large central stations.

The most economical size and number for an assumed set of operating conditions can be determined only by considering the various influencing factors, such as load characteristics of the individual boiler units themselves, first cost, maintenance, and nature of the total plant load. High peak loads of short duration usually warrant the installation of a few units with heavy overload characteristics, while uniform loads are handled more economically with a large number of units operating near their point of maximum thermal efficiency. *It should be borne in mind that extremely high boiler ratings are obtainable only with the best of furnace constructions, scale-free feedwater and first class supervision, and since these conditions are seldom found in any but the largest plants, it is better, as a general rule, to err in installing too many units than in attempting to operate a smaller number of undersized units at continuous overloads.* A study of a number of the latest installations shows that fewer boilers are being installed than formerly for the same conditions, owing to better furnace construction, improved methods of handling fuel, and provisions for feedwater purification. Because of the great number of variables entering into the problem of determining the number and size of boiler units for a given maximum capacity, general rules are without purpose except for very rough approximations. In central station practice of a decade ago, boiler units of more than 10,000 sq. ft. of heating surface each were exceptional, and the ratios of water-heating surface to kilowatts of rated

generator capacity ranged from 3 to 1, to 5 to 1; but, in the very latest designs, individual boiler units of less than 15,000 sq. ft. of heating surface are seldom installed, and the ratios of water-heating surface to kilowatts of installed generator capacity range from 1.2 to 1, to 2.5 to 1.

86. Boiler Accessories. — All steam boilers must be protected by safety and relief valves, and by such indicating and controlling devices as will insure their safe operation. Such devices, including appliances or fittings which are either intimately connected with the boiler structure or with the work of boiler operation and maintenance, are commonly designated as **boiler accessories.** The design and installation of the more important safety devices are controlled by law and insurance requirements. Considering the fact that the A.S.M.E. Boiler Code has already been adopted by a number of states and no doubt will eventually supersede all others, except perhaps those under federal control, only the devices and installations recommended by the Code will be discussed.

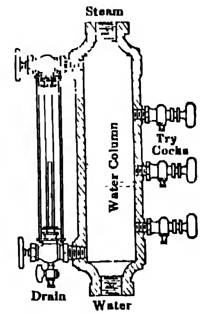

FIG. 69. Simple Water Column.

Safety Valves. See paragraph 316.

87. Water Gages. — The water level in a boiler is usually indicated either by a **gage glass,** by **try cocks,** or both, connected directly to the boiler as in Fig. 1, or to a **water column** or **combination** as in Fig. 69. Water gages and water columns should be so located that the normal water level is near the center of the gage or column. The upper try cock should be located at the highest permissible water level and the lower one at the lowest level, and the position of the middle cock should correspond to normal water level. In the simple water column illustrated in Fig. 69, the gage glass connections are fitted with simple stop valves for shutting off the steam or water in case the glass is broken. In high boilers the water-gage valves are usually of the quick-closing type, Fig. 70, which may be operated from the boiler room level by means of a chain attached to the valve stem. Try cocks for high boilers are similarly operated by chains and are automatic in closing (see Fig. 71). Certain types of automatic water-gage valves for automatically cutting off the water or steam supply when the gage glass breaks are permitted by the A.S.M.E. Code, but their use is not common. Water columns fitted with hand-operated drain cocks should be "blown out" periodically to remove any sedi-

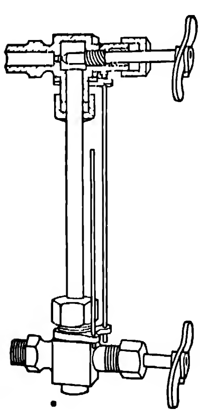
FIG. 70. Quick Closing Water Gage Valves.

mental deposits. By connecting the drain opening directly with lower column connection, the drain cock may be dispensed with and sediment will not lodge in the bottom of the glass. This system of drainage is a common practice. Water columns are frequently fitted with float-controlled whistles, as illustrated in Fig. 72. These alarms automatically give a warning signal when the water level is too high or too low. Instead of a whistle, the floats may actuate an electric circuit which in turn may light a lamp, ring a bell or buzzer, or record the time of opening on a chart. Water columns are usually connected to the boiler without shut-off valves, but if such valves are used the A.S.M.E. Code prescribes that "they shall be either outside-screw and yoke-gate valves (see paragraph 310), or stop cocks which have levers permanently fastened thereto, and such valves or cocks shall be locked or sealed *open*." The Code also stipulates that the piping between water column and the boiler shall have no outlet connections except for damper regulator, feedwater regulator, drains or steam gages.

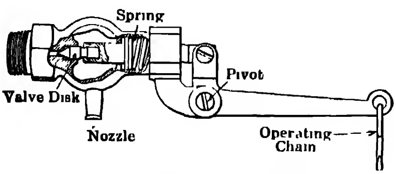

FIG. 71. Self-closing Gage Cock.

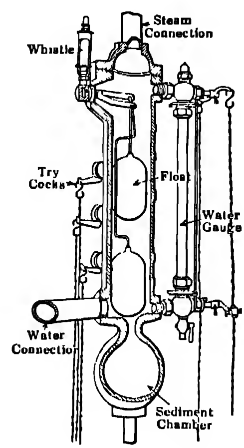

FIG. 72 Combined Water Column and High and Low Water Alarm.

88. Fusible or Safety Plugs. — Fusible or safety plugs, as illustrated in Fig. 73, are brass plugs provided with a fusible metal core. They are inserted in the shell or tubes at the lowest permissible water line. When they are covered by water the heat is conducted away sufficiently fast to keep the temperature below the fusing point, but when they are uncovered the low conductivity of the steam prevents the rapid withdrawal of heat, whereupon the alloy melts and the blast of escaping steam gives warning. The melting point of fusible metals being sometimes uncertain, plugs occasionally blow out without apparent cause and at other times fail to act when shell is overheated. Fusible plugs may be attached directly to the boiler heating surface or in a fitting attached to the boiler, so that the flow of water or steam may be shut off when the plug melts.

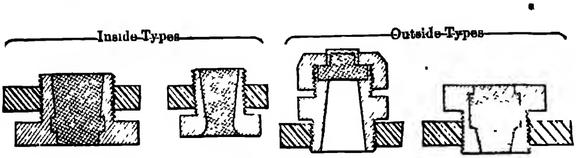

FIG. 73. Types of Fusible Plugs.

The A.S.M.E. Code considers only the directly attached plugs and recommends where they should be inserted in the various types of boilers.

89. Blow-offs. — Boilers must be provided with blow-off pipes for draining off the water and for discharging sediment and scale-forming material. The "bottom blow" is ordinarily an extra heavy pipe of suitable diameter connected to the mud drum or to the lowest part of the

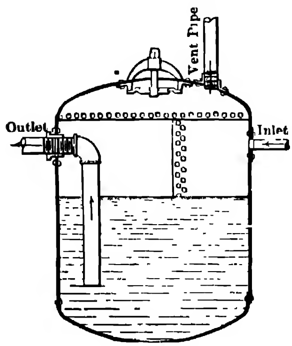

FIG. 74. Blow-off Tank and Connections.

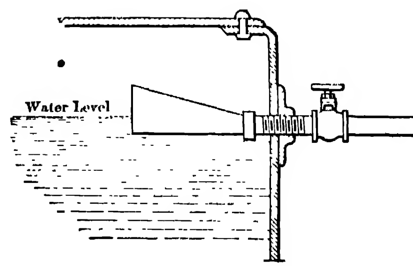

FIG. 75. Surface Blow.

boiler and fitted with two valves or cocks, or a valve and cock (see paragraph 315). The generally approved method of arranging the blow-off pipe for a return-tubular boiler is shown in Fig. 117. This method of protecting the pipe from the direct action of the heated gases by means of a V-shaped brick pier permits easy examinations of the blow-off through the cleaning door in the rear wall of the setting. Where boilers are arranged in batteries, the battery may have a common outlet for the blow-off pipes. The blow-off pipes are frequently discharged into the open air, but this is not permissible in large cities, nor is it lawful to blow directly into the sewer. In this case, the water and sediment may be discharged into a blow-off tank, Fig. 74, and permitted to cool before delivery to the sewer.

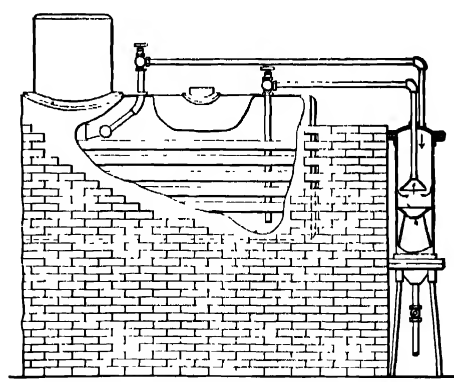

FIG. 76. Skimmer — Floating Type.

Surface blows are occasionally installed to remove scum, grease, and floating or suspended particles of dirt in small plants where the water is

particularly bad. The bell-mouthed shape shown in Fig. 75 permits the skimmer to accommodate itself to varying water levels. Skimmers are sometimes provided with a flexible jointed float, Fig. 76.

90. Damper Regulators. — In hand-fired natural-draft furnaces, the amount of air admitted to the furnace and amount of flue gas passing to the stack is usually controlled by the boiler or stack damper. This control may be manual or automatic. There are countless automatic damper regulators on the market, and practically all low-pressure heating boilers are equipped with such appliances. In high-pressure installations, however, manual control is the more common and automatic control the exception. The majority of damper regulators for hand-fired, natural-draft boilers depend upon variation in steam pressure as the primary control. The difference in pressure may act directly upon a steam piston to which the damper is connected by suitable linkage, or it may actuate a relay system so that the movement of the damper is effected by compressed air, by water under pressure, or by an electric motor. Low-pressure regulators consist usually of a flat or sylphon diaphragm with direct boiler pressure on one side and the damper linkage on the other. High-pressure regulators are either of the direct-pressure type or of the relay type.

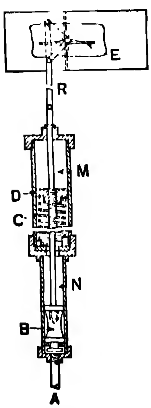

FIG. 77. Typical Steam-actuated Damper Regulator.

Figure 77 shows a section through the simplest form of the high-pressure direct-steam-actuated type. The device is connected directly to the boiler by pipe A. The pressure on piston B is balanced by spring C under normal conditions of operation. Any variation from the normal steam pressure will cause the rod R to move up or down so that the damper is opened or closed in proportion to the change in pressure. The chamber N is separate from chamber M so that steam cannot come into contact with the spring. Piston D acts as a guide and prevents sudden movements of the main actuating piston.

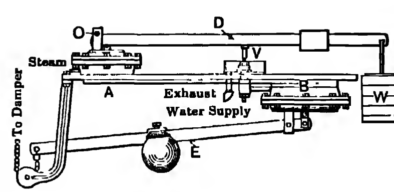

FIG. 78. Typical Hydraulic Damper Regulator.

Figure 78 illustrates a typical mechanism of the indirect type. Full boiler pressure acting at all times on the diaphragm A raises or lowers a weight W attached to arm D according to the increase or decrease of pressure. Arm D actuates a small valve V, which controls a supply of water under pressure to chamber B. The water pressure acts on the

diaphragm in chamber B, which in turn moves the damper through the agency of weighed lever E.

Dampers similar to those just described are satisfactory where there are no sudden variations in steam pressure, but, where such changes occur, the damper is apt to be shifted from "wide open" to "shut," resulting in a continued "hunting" action between fluctuation in steam pressure and damper movement. In the latest designs this hunting action is avoided by moving the damper in graduated steps and by effecting a delayed action between each step. Among the latter may be mentioned the **McDonaugh, Ruggles-Klingemann** and the **National.** When correctly adjusted and given proper attention, automatic dampers of the graduated-step type result in satisfactory fuel savings over hand control.

In the modern stoker-fired plants with natural or induced draft, the movement of the damper is coordinated with the control of the stoker and fan engines. See paragraph 161.

91. Soot Blowers, Tube Cleaners, etc. — Aside from the assurance against burning out of tubes due to the accumulation of scale, the maintenance of clean heating surfaces is one of the most important problems in connection with recent developments toward higher boiler ratings and in the operation of large boiler units. Efficiency and capacity depend to a greater extent upon cleanliness (both internal and external) of the heating surfaces than is ordinarily realized. Soot is an excellent heat-insulating material, and consequently any appreciable deposit on the heating surfaces will reduce the rate of heat absorption and result in high flue-gas temperatures. The gain effected in economy and capacity by the removal of soot varies with depth, extent, and nature of the deposit and with the rate of driving. No modern plant is operated without periodically removing this deposit.

Surfaces exposed to the action of the products of combustion are customarily freed from soot and clinkers by steam lances, soot blowers incorporated within the setting, brushes, scrapers, and similar appliances. Light, flocculent soot is conveniently removed at regular intervals by means of a hand-operated steam lance with which all surfaces are reached and swept clean. Under certain conditions more economical results are obtained by permanently installed soot blowers. (See Figs. 79 and 80.) These consist of a series of pipes and nozzles, the latter stationary or revolving, located so that all parts of the heating surface subjected to soot deposit may be swept with a jet of steam. In the older designs, individual hand-controlled valves are placed in the pipe branches leading to the blower element; in some of the more recent designs the valves are incorporated in the head of the element so that manipulation of the chain opens and closes the valves. Electric control and electric operation of

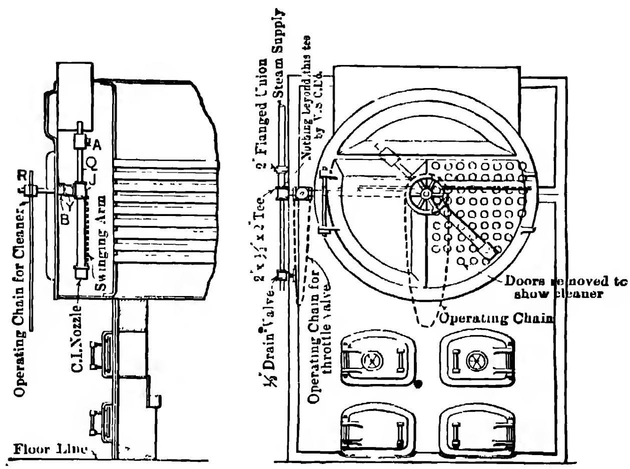

FIG. 79. Soot Blower — Return Tubular Boiler.

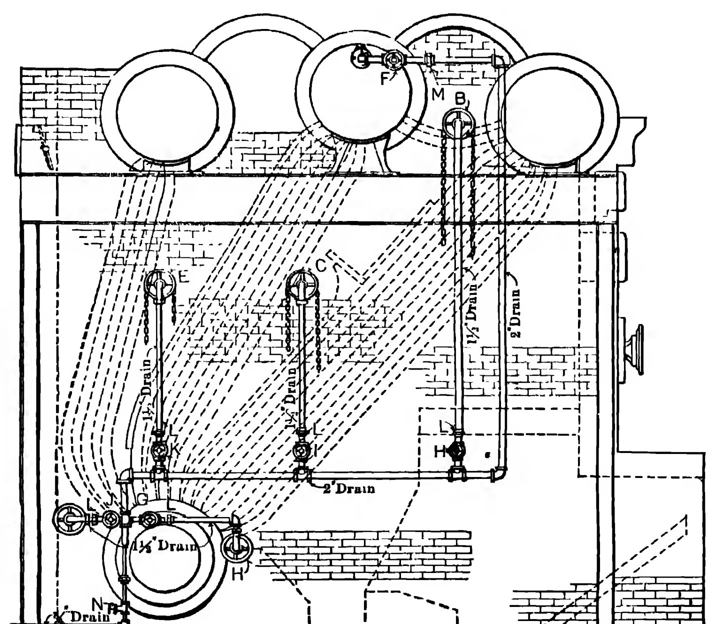

FIG. 80. Soot Blower — Individual Branch Pipe Control

the individual heads by means of small motors is also in evidence in some of our latest stations. A soot blower is considerably superior to a hand lance from a safety standpoint. For steam consumption of soot blowers, see paragraph 60. With certain grades of coal under heavy furnace capacity, the particles of ash and slag carried along with the products of combustion are in a plastic state and adhere to the two or three lower tubes. The accumulation may eventually result in a complete choking up of the gas passages. Blowing by hand lances and machine blowing devices will not remove the accumulation, and dislodging the deposit with pokers, after the furnace has been partially cooled, appears to be the only practical solution of the problem.

The question of preventing the formation of scale by purification of the feedwater and the loss in heat transmission due to scale deposit is treated at length in Chapter XIII. In the average plant, furnished with commercially good feedwater, it is a common practice to allow scale to deposit for a limited period of time and then remove it mechanically by tube cleaners and scrapers. The principles of construction of these devices vary widely according to the types of boilers in which they are used, and depend upon the nature of the duty which they must perform. Mechanical tube cleaners may be conveniently divided into two classes:

1. Those which loosen the scale by a series of rapid hammer blows, Fig. 81.

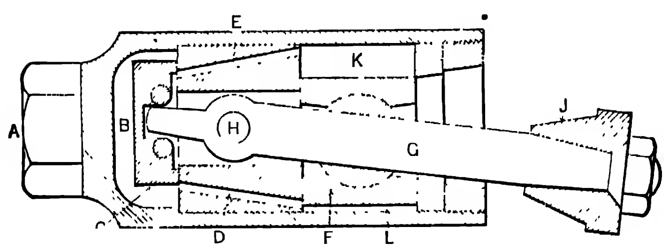

Fig. 81. Mechanical Tube Cleaner — Hammer Type.

2. Those which cut out the scale by a revolving tool, Fig. 82.

The **hammer** device is applicable to either the water or fire-tube type of boiler, but the **revolving cutter** is applicable to the water-tube only. Compressed air, water under pressure, or steam may be used as the motive power for turbine cleaners, and steam or air for hammer cleaners. Water is ordinarily the most convenient for the turbine type, but air increases the capacity of the cutter and is finding favor with many engineers.

Referring to Fig. 81, the hammer head *J* is given a rapid motion, which may reach 1500 vibrations per minute, and subjects the tube to repeated

shocks, thereby cracking the brittle scale and jarring it loose from the water surface of the tube. The cleaner head is attached to a flexible pipe of sufficient length to enable it to be pushed from one end to the other. Unless carefully manipulated, the hammer is apt to injure the tube by swaging it to a larger diameter and the vibrations may cause leaks where the tubes are expanded into the tube sheets.

Turbine cutters are made in many designs, one of which is shown in Fig. 82. The particular device illustrated in Fig. 82 is of the hydraulically driven type. A high speed of rotation is imparted to the cutter head by a small paddle wheel or turbine located as indicated. The cutters chip the scale into small pieces, and the stream of air flowing from the turbine envelops the cutters, keeps their edges cool, and washes away the scale as fast as it is detached. Different styles of cutter wheels are furnished with each cleaner so as to adapt the device to all kinds of scale formations. In well managed plants using raw feedwater, scale is not permitted to deposit to a thickness greater than 1/32 to 1/16 of an inch. Small-sized tube cleaners for superheaters are frequently operated by compressed air.

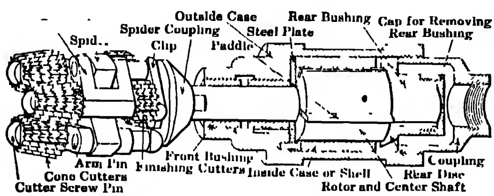

FIG. 82. Mechanical Tube Cleaner — Cutter Type.

Soot Removal from Fire-tube Boilers: Power, Aug. 27, 1918, p. 305.
Economies of Mechanical Soot Blowers: Power Plant Engrg., Nov. 1, 1923, p. 1070.

PROBLEMS

1. Given: initial pressure 115 lb. abs.; barometer 29.92 in.; quality 98 per cent; feedwater, 82 deg. fahr. Required b. hp. necessary to furnish a 50-hp. engine with steam; engine uses 45 lb. per i.hp.-hr.

2. A 30,000-kw. steam turbine and auxiliaries require 12 lb. steam per kw.-hr. at rated load; initial pressure 260 lb. abs.; superheat 250 deg. fahr.; feedwater 180 deg. fahr. Required the b. hp. necessary to furnish the turbine and auxiliaries with steam. If the boilers are operated at 250 per cent rating when supplying the turbine and auxiliaries, required the ratio of kw. turbine rating to b. hp. rating.

3. A boiler evaporates 90,000 lb. of water per hr. from a feed temperature of 210 deg. fahr. to steam at 300 lb. absolute pressure and 200 deg. superheat. If the boiler is being forced to 200 per cent rating when evaporating this amount of water, required the extent of heating surface, assuming that the normal rating corresponds to an evaporation of 3.45 lb. water from and at 212 deg. fahr. per sq. ft. of heating surface. Allowing 10 sq. ft. of heating surface per rated b. hp., required the boiler rating.

4. Determine the factor of evaporation for Problems 1 and 2.

5. The following data were taken from a boiler test:

Heating surface, 8000 sq. ft.; grate surface, 160 sq. ft.; furnace vol., 1600 cu. ft.

Coal analysis (as fired): moisture 8 per cent; ash 12 per cent; B.t.u. per lb. 12,100.

Weight per hr.; water fed to boiler, 32,000 lb.; coal 4000 lb.; dry refuse removed from ashpit, 720 lb.

Temperatures: flue gas, 480 deg. fahr.; feedwater, 160 deg. fahr ; boiler room, 80 deg. fahr.; relative humidity, 50 per cent.

Pressures: steam pressure, 125-lb. gage; barometer, 29.0 in., superheat, 100 deg. fahr. Required:

a. Factor of evaporation.

b. B. hp. developed.

c. Per cent of builder's rating developed (builder's rating = 10 sq. ft. of heating surface per b. hp.

d. Evaporation lb. per lb. of coal as fired:

(1) Actual
(2) Equivalent

e. Evaporation lb. per lb. of dry coal:

(1) Actual
(2) Equivalent

f. Evaporation lb. per lb of combustible:

(1) Actual
(2) Equivalent

g. Equivalent evaporation lb. per lb. of combustible burned.

h. Evaporation lb. per sq. ft. of heating surface.

(1) Actual
(2) Equivalent
(3) No. of 1000 B.t.u. absorbed per hr. per sq. ft of heating surface.

i. Combustion space per lb. coal as fired per hr , cu. ft.

j. Heat value of the combustible as fired, B.t.u. per lb.

k. Heat value of the combustible as burned.

l. Efficiency of the boiler, furnace, superheater, and grate, per cent.

m. Efficiency on the combustible basis, per cent.

6. The following additional data were taken during the test outlined in Problem 5: flue-gas analysis, per cent by volume: CO_2, 14 19; CO, 1.12, O, 3.54; N, 80.85; heat value of combustible in refuse, 13,500 B.t.u. per lb.

Ultimate analysis of coal as fired, per cent by weight:

Carbon 66, hydrogen 5, nitrogen 1, oxygen 8, moisture 8, ash 12.

Calculate on the coal as fired basis:

(1) Complete heat balance.
(2) Inherent losses.
(3) Per cent of available heat utilized.

7. If the fuel, analysis as in Problem 22, cost $6.00 per ton, determine the fuel cost of evaporating 1000 lb. water from and at 212 deg. fahr.

8. A test of an oil-fired furnace gave an actual evaporation of 13 lb. water per lb. of oil with boiler, furnace, and superheater efficiency of 80 per cent; boiler pressure 200 lb. abs., superheat 100 deg. fahr., feedwater temperature 162 deg. fahr. Required the heating value of the fuel.

CHAPTER V

SUPERHEATERS

92. Advantages of Superheating. — That superheated steam results in ultimate plant economy is evidenced by the fact that the largest and most economical plants in the world are equipped with superheaters. A limited amount of superheat can be used with practically any equipment, and it effects ultimate economy in nearly all cases. Higher superheats require specially designed equipment. Practically all modern central turbo-generator stations and large isolated piston-engine plants are designed for superheated steam. No general rules can be drawn as to the extent of the saving made, because of the great number of variable factors entering into the problem. Each installation must be considered by itself and due consideration given to such items as the type and size of prime movers, character of service, nature and cost of fuel, piping, first cost, upkeep and the like. The logical procedure is to determine the saving in fuel regardless of other factors and then deduct the extra expense due to first cost and upkeep. The resulting net gain or loss will show whether or not the use of superheat is advisable.

Theoretically, all types of steam-driven prime movers show increased heat efficiency with superheated steam, but the gain is usually less than that actually realized in the commercial mechanism. Aside from the gain in the prime mover, there is the possible added efficiency in the boiler plant. It is true that the heat required to superheat steam is furnished by the fuel, and when a definite weight is superheated an added amount of fuel must be burned; but with a properly designed superheater integral with the boiler, the overall efficiency of boiler and superheater is usually somewhat higher than if saturated steam alone were generated, so that the added amount of fuel is less than the heat gained by the steam. In addition to the thermal gain in the prime mover and boiler, there may be a reduction in heat losses in the piping system because smaller pipes may be used and because superheated steam gives up heat less rapidly than does wet steam.[1] Furthermore, the increased economy of the prime mover may permit a reduction in the size of boilers, condensers, and other auxiliary apparatus.

At high temperatures superheated steam behaves like a gas and is,

[1] *Lower Line Losses with Superheated Steam*: Power, Aug. 7, 1923, p. 233.

therefore, in a far more stable condition than saturated steam. Considerable heat may be abstracted without producing liquefaction, whereas the slightest absorption of heat from saturated steam results in condensation. If superheat is high enough to supply not only the heat absorbed by the cylinder walls but also the heat equivalent of the work done during expansion, the steam will be dry and saturated at release.

According to Ripper ("Steam Engine Theory," p. 155), this is the condition of maximum efficiency in a single cylinder, but several tests of reciprocating steam engines conducted by the author gave maximum efficiency when the exhaust showed superheat varying from 10 to 25 deg. fahr. Long observations on steam engines show that in order to obtain dry steam at release, the superheat at cut-off must be between 1 1/2 and 2 1/2 times the total temperature drop which would occur if the engine were operated with saturated steam, depending upon the initial condition of the steam and the ratio of expansion. Under "temperature drop," is understood the temperature difference between the live steam and the exhaust steam. The earlier the cut-off, the higher the degree of superheat required. A superheat of not less than 250 deg. fahr. at admission is necessary to secure dry steam at release in the average single cylinder cutting off at one-fourth stroke and with boiler pressure of 100 lb. gage. Small steam turbines for auxiliary drives frequently show superheat in the exhaust when the initial superheat is only 100 deg. fahr. There will be a reduction of approximately 1 per cent in cylinder condensation for 7.5 to 10 deg. of superheat. In Europe it is common practice to superheat the steam between each stage of compound and triple expansion engines, but this practice is not generally followed in America because of the complications involved. A high-pressure turbine designed for 1200 lb. gage pressure and initial temperature of 750 deg. fahr. is now (1926) being operated in the Weymouth station of the Edison Electric Illuminating Co., Boston, Mass. The exhaust from this unit is to be reheated to 750 deg. and discharged at 300 lb. pressure into the steam mains which supply the large turbines at the station.

The water rate of the steam turbine is decreased by superheating, but to a less extent than that of the piston engine. Theoretically, the improvement in steam economy is the same for both types of prime movers, pressure and temperature ranges being the same in each case, but in actual practice the gain is more pronounced with the piston engine. This is due to the fact that with reciprocating engines the live steam entering the cylinder comes in contact with cylinder walls which were previously cooled as a result of heat being abstracted in the re-evaporation of moisture in the exhaust steam. This results in condensation losses when the live steam is not superheated. With superheated steam, the condensation

and re-evaporation losses are eliminated, or at least considerably reduced. In general, the less economical the steam motor, the more is the gain effected by superheating. Aside from the gain in heat efficiency, the use of superheated steam benefits the turbine by reducing erosion of the blades and by lowering skin friction and windage. The fact that nearly all modern steam turbine plants are operated with superheated steam is evidence that superheating results in ultimate plant economy. Where it becomes necessary, particularly with old boilers, to reduce the operating pressure, with a consequent decrease in plant capacity, the application of superheat to such boilers will enable them to meet the power demands of the plant at the reduced pressure. In most cases, superheating will provide additional reserve power over that of the plant before the pressure was reduced.

Industrial Uses of Superheated Steam: Trans. A.S.H. & V.E., Vol. 25, 1919, p. 365.

93. Economy of Superheat. — Many comparative tests of engines and turbines using saturated and superheated steam, under varying conditions of pressure and temperature, have been made during the past few years, showing in all cases decreased steam consumption due to superheat. Substantial ultimate gains are effected with moderately superheated steam, but in view of the still greater economies possible with highly superheated steam, with little additional cost for equipment, it is advisable to use the highest superheat which plant conditions permit. In many new plants, particularly those of larger capacity, 250 to 300 deg. fahr. of superheat are being used successfully. The first cost is not excessive; repairs are moderate; and the life of the installation is all that can be desired.

As far as steam consumption per hp-hr. is concerned, superheating usually increases the economy of the piston engine from 5 to 15 per cent and in some instances as much as 40, the latter figure referring to the more wasteful types. A fair estimate of the comparative ranges in steam consumption of various types of prime movers using saturated steam, and steam superheated 100 and 200 deg. fahr., is given in the following table:

Type of Engine	Economy in Steam Consumption (Per Cent) Over Saturated Steam	
	100 Deg. Superheat	200 Deg. Superheat
Simple, non-condensing	15–30	20–38
Compound, non-condensing	12–22	18–32
condensing	10–18	16–28
Triple, condensing	8–15	12–22
Turbine, non-condensing	10–18	16–28
condensing	8–12	14–20

European builders guarantee steam consumption with highly superheated steam (total temperatures 750 to 850 deg. fahr.), as follows:

	Lb. per I.hp.-hr.
Single-cylinder condensing engines (uniflow) . . .	8.5
Single-cylinder non-condensing engines (uniflow)	12 0
Compound condensing engines (locomobile) .	8 0
Compound non-condensing engines (locomobile) .	10 5

An exceptionally low steam consumption is credited to a tandem compound using steam superheated to 815 deg. fahr. at an initial pressure of 794 lb. abs. When exhausting against an abs. back pressure of 0.7 lb., the steam consumption was 7.12 lb. per i.hp-hr., corresponding to a thermal efficiency of approximately 30 per cent. (*Power*, Feb. 7, 1922, p. 219.)

In high-pressure steam turbines, the water rate is improved approximately 1 per cent for every 8 to 12 deg. fahr. superheat, the higher rate holding for about 50 deg. superheat and the lower for about 200 deg. It is difficult to estimate the actual gain in heat economy due to superheating in very large turbines, since they are not designed for saturated steam and tests with the latter do not offer a true comparison. In a general way the average reduction in steam consumption for these large units is about 1 per cent for every 10 deg. fahr. increase in superheat.

In comparing the performance of engines and turbines using saturated steam, it is advisable to base all results on the heat consumed per unit output rather than on the steam consumption, since the latter is apt to give a false idea of the relative economies. The real measure of economy is the cost of producing power, taking into consideration all charges, fixed and operating, and the next best is the coal consumption per unit output; but as a means of comparing the motors only, the heat consumption per unit output is very satisfactory.

See paragraph 186 for the influence of superheat on the economy of reciprocating engines, and paragraph 214 for the influence on steam turbines.

94. Limit of Superheat. — In this country, steam temperatures of 600 deg. to 650 deg. fahr. are common on locomotives. These temperatures are being used also in many stationary plants of large capacity. In plants of moderate size, and especially those which are converted from saturated to superheated steam, it is advisable to use the maximum superheat which existing conditions allow, and which good engineering practice dictates as being safe. While, heretofore, moderate superheat has been considered satisfactory, the cost of fuel necessitates the utmost economy, and higher superheat should be used wherever possible to effect

economy. In Europe, few if any plants are installed without superheaters, and 700 deg. is a common temperature, with a maximum of about 850 deg. There is no particular mechanical difficulty in designing power plant apparatus to withstand temperatures as high as 850 deg. fahr., and for industrial purposes still higher steam temperatures are often used. In general, it may be said that each case should be considered in all its details before a decision is reached relative to the most advantageous superheat temperature to be used.

Experience has shown that with engines of ordinary design, slide-valve and Corliss, the temperature at the throttle should not exceed 500 deg. fahr. This corresponds to a superheat of 160 deg. with steam at 100 lb. gage pressure, and 130 deg. at 150 lb. This degree of superheat insures practically dry steam at cut-off in the better grade of engines. Just how far superheating can be carried with a given engine of ordinary construction can be determined by experiment only, but a temperature of 500 deg. is probably an outside figure and 450 deg. a good average. Higher temperatures are apt to interfere with lubrication and sometimes cause warping of the valves. With temperatures below 450 deg., no difficulties are ordinarily encountered.

With highly superheated steam involving temperatures of 600 deg. fahr. or more, the poppet-valve type of engine is ordinarily employed, though balanced piston and specially designed Corliss valves are not uncommon. The poppet valve is not distorted by heat and requires no lubrication. In Europe these engines have been brought to a high state of efficiency, but have not been generally adopted in this country. The steam end of the composite gas-steam engines at the Ford Motor Company's plant, Detroit, are of Corliss valve design and though the steam at admission has a temperature of 700 deg. fahr. no difficulty is experienced with lubrication.

Owing to the absence of rubbing parts in contact with the steam, and because the casing is not subjected alternately to high and low temperatures, steam turbines may be designed to operate successfully with temperatures up to 850 deg. fahr., though temperatures above 700 deg. are exceptional. The latest steam turbine installations in this country are designed for temperatures of 750 deg. fahr.

95. Types of Superheaters. — Superheaters are broadly classified as *convection* superheaters and *radiant* superheaters, according to the source of heat. The former are usually placed in the boiler gas passages where the heat is transmitted mainly by convection, and the latter in the walls of the furnace where the heat transmission is by radiation. The convection type, which is by far the more common, may be grouped into two classes:

(*a*) Independently fired, and

(*b*) Integrally built superheaters, which are installed within the boiler setting.

The **independently fired** superheater is not widely used, owing mainly to the lower economy effected in fuel when compared with that obtained through the use of the integrally built type. However, there are many places where it is convenient and advisable to use thein dependently fired superheater. For example, it may be desirable to have a small amount of steam superheated to a high degree. Where waste gases of high temperature are available, the independently fired superheater may be used to advantage. In general, the independently fired superheater is confined to special and limited uses. The **integrally built** superheater may be located in the furnace, as in Fig. 94, at the end of the heating surface as in Fig. 97, or at some intermediate point, as in Figs. 86 and 89. Since the absorption of heat depends chiefly upon the average temperature difference between the gases and the steam and the extent of superheating surface, the required degree of superheat may be obtained from a small extent of heating surface in the furnace, a large amount in the rear of the heating surface, or a proportionate amount in intermediate locations. In a general sense, the sum of the boiler heating surface and superheating surface per b.hp. is practically the same for any degree of superheat. The cost of a superheated steam boiler is approximately equal to that of a saturated steam boiler, since the superheated plant has less steam to generate. The requirements of a successful superheater are: (1) security of operation, or minimum danger of overheating; (2) uniform superheat at varying ratings of the boiler; (3) economical use of heat applied; (4) provision for free expansion; (5) provision for cutting out superheater without interfering with the operation of the plant; (6) provision for keeping tubes free from soot and scale.

Superheaters may be **separately fired** or **indirectly fired.** The advantages of the separately fired superheater are as follows: (1) The degree of superheat may be varied independently of the performance of the boiler. (2) The superheater may be placed at any desired point. (3) Repairs are readily made without shutting down the boiler.

The following are some of the disadvantages: (1) Superheater requires separate attention. (2) Saturated steam only can be furnished to the prime movers in case of a breakdown of the superheater. This can be minimized by arranging the superheater in two sections, so that when one section is shut down the entire amount of steam can be sent to the other section. The superheat is then somewhat lower, and the pressure drop through the superheater correspondingly higher. (3) Extra piping is required. (4) Extra space is required.

The indirectly fired superheater arranged in the boiler setting has the advantage of: (1) lower first cost; (2) higher operating efficiency; (3) minimum attention, (4) minimum space requirements.

Until recently, superheaters integral with boilers were located in the path of the flue gases after they had passed a considerable amount of heating surface and had given up the greater part of their heat. With superheaters so located, the steam temperature increases appreciably with the increased load. Figure 102 shows the relation between superheat and boiler rating with a superheater located between the first and second passes of a standard water-tube boiler. With the development of the large power stations and large generating units, more uniform superheat has become a necessity, and superheater designs have undergone new developments. Now, more than ever before, it is recognized that as long as steam is flowing through a superheater, its elements are protected from overheating by the cooling action of the steam, and the only time when they are endangered is during the firing-up period. The general practice in larger power stations is to keep boilers on the line all the time. They are shut down only for repairs or cleaning, so that the firing-up periods are less frequent. In modern power plants with large boilers, the superheaters are, therefore, located in a hotter gas zone. In order to give uniform superheat at various boiler ratings, the capacity of a superheater must vary with that of the boiler. The ideal method of effecting this result would be to distribute the superheating surface throughout that portion of the gas path where heat is given up to the water in the boiler. This, however, is not done, for practical reasons, but a superheater located between the boiler tubes, which generate most of the steam, gives a practically constant steam temperature. Superheaters having this characteristic are illustrated by Figs. 86 and 89.

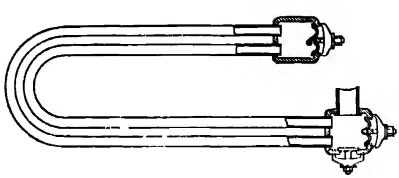

FIG. 83. Babcock and Wilcox Superheater Assembly.

In a number of recent installations, the superheater is placed directly in the furnace, and absorbs the heat by radiation (Fig. 94). With such location the temperature of the steam drops off with increase in boiler rating. By placing part of the superheating surface in the path of the gases, and part in the furnace, the rising characteristic of the convection type and the drooping characteristic of the radiant type will produce practically constant superheat at all loads.

Figure 83 gives the general details of a **Babcock & Wilcox** superheater, and Fig. 84 shows the application of superheating coils to a longitudinal-drum Babcock and Wilcox boiler, illustrating the usual location of the

indirectly fired type. The superheater consists of two transverse, square wrought-steel manifolds into which two sets of 2-in. cold-drawn seamless steel tubes bent in a " U " shape are expanded. The tubes ordinarily are arranged in groups of four. Saturated steam flows from the dry pipe, located within the drums, to the upper manifold. The latter is divided into as many sections as there are drums, so as to avoid expansion strain. From the upper manifold, the steam passes through the U-shaped tubes to the lower one (which is continuous) and thence to a cast-steel " superheater center" fitting supported over the drum. The " superheater center " fitting is provided with a superheated steam outlet and an extra opening for the reception of the superheater safety valve. This safety valve is furnished as a part of the regular equipment and is set 2 lb. lower than the safety valves of the boiler. This is essential in order to provide a flow of steam through the superheater and to prevent any overheating of the latter in case the load should be suddenly thrown off the boiler. A small pipe connects the center fitting with the saturated steam space in the drum and is for the purpose of equalizing the pressure when the discharge from the superheater is closed. While a flooding device is not necessary, its use is frequently recommended by the Babcock & Wilcox Company. This consists essentially of a small pipe which connects the lower manifold with the water space of the boiler and by means of which the superheater may be flooded. Any steam formed in the superheater tubes is returned to the boiler drum through the collecting pipe, which, when the superheater is at work, conveys saturated steam into the upper manifold. When steam pressure has been attained, the superheater is thrown into action by draining the water away from the manifolds and opening the superheaterstop valve. With the proportion of superheating surface to boiler surface ordinarily adopted, the steam is superheated from 100 to 150 deg. fahr.

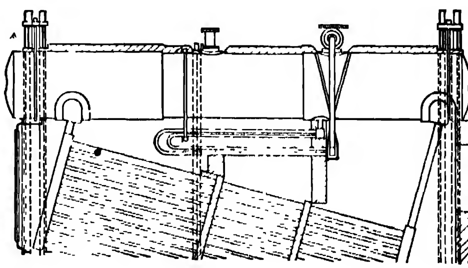

FIG. 84. Babcock and Wilcox Superheater — Usual Location.

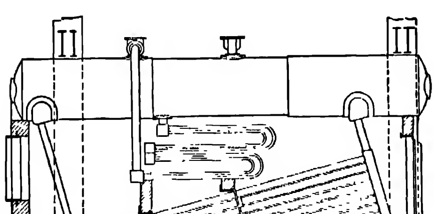

FIG. 85. Babcock and Wilcox Superheater — Double Deck (Usual Location).

The tubes of the Babcock & Wilcox superheaters, as applied to Stirling boilers, are ordinarily equipped with ferrules or cores, as conditions warrant, to give a proper ratio of tube cross-sectional area to header area. By the use of this construction, it is assured that all tubes carry their proper proportion of the total amount of steam passing through the superheater, and the danger of warping or burning of any tubes due to being by-passed is obviated.

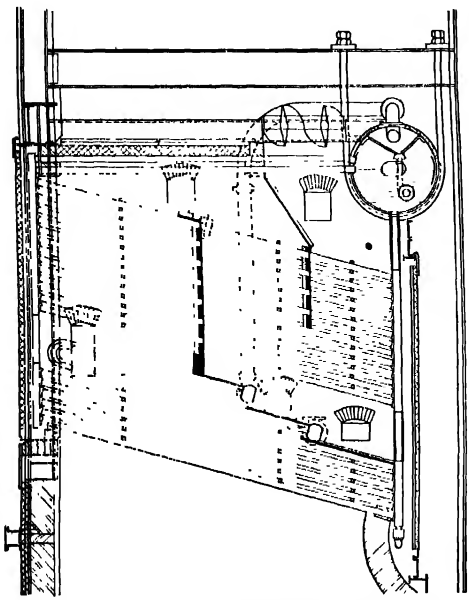

FIG. 86. Babcock and Wilcox Superheater — Located Between Tube Sections.

With single inlet and outlet superheaters, one end of each superheater header is welded closed. To the other end there is attached a wrought-steel flange.

Figure 42 shows a section through a **duplex superheater** as installed in the 1200-lb. boiler at Calumet. The primary superheater superheats the steam generated by the boiler, while the secondary superheater reheats the steam exhausted at 300-lb. gage from the high-pressure turbine. The secondary superheater incloses the primary as indicated.

Figure 87 gives a side elevation of one section of an **Elesco superheater** as installed in the Springfield boilers at the Hell Gate Station. The headers are made of extra heavy pipe and are so located that the joints between headers and elements are placed in a comparatively cool gas zone, and are easily accessible for inspection and repairs. The elements consist of cold-drawn seamless tubing of small diameter, so that no cores are required. The elements are fastened to the header, by means of a detachable metal-to-metal

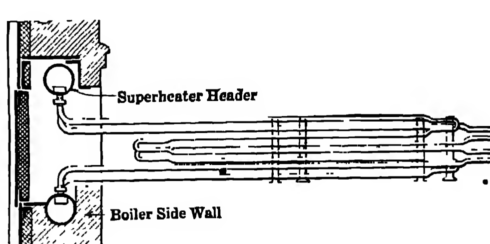

FIG. 87. Assembly of Elesco Superheater — Side Elevation.

ball clamp joint, and no welding, rolling, or gaskets of any kind are applied. The clamp studs are made of special heat-treated alloy steel with a minimum tensile strength of 100,000 lb. per sq. in. Figure 88 shows a section through the header and joints.

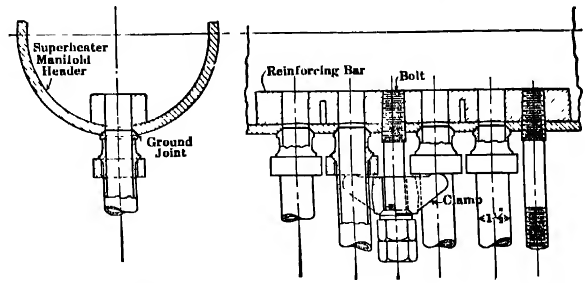

Fig. 88. Method of Securing Tubes to Header — Elesco Superheater.

The elements have forged return bends, an important feature of these superheaters, since it permits placing of elements closer together and better utilizes the space available for the superheater. Owing to the sharp turn, a better mixing of the steam is obtained, resulting in a more uniform heating.

Any slugs of water carried over by the steam are broken up in the return bend, and become easily evaporated. These return bends are made on the ends of the units from the metal of the pipe itself, by a special mechanical forging process without the use of electrical or acetylene welding. By means of the return bends, long units can be manufactured, decreasing the number of joints in the header.

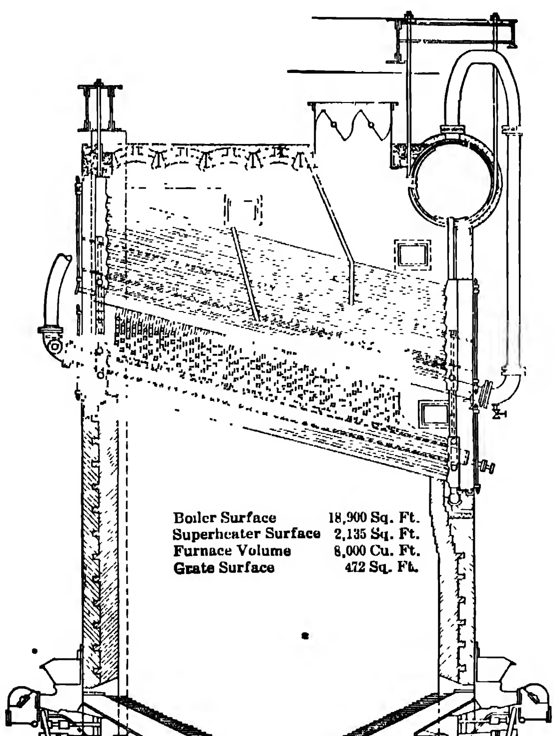

Fig. 89. Elesco Superheater Installation — Hell Gate Station.

Figures 89 and 90 are examples of two installations in accordance with modern practice in superheater design. Figure 89 shows a large cross-

The tubes of the Babcock & Wilcox superheaters, as applied to Stirling boilers, are ordinarily equipped with ferrules or cores, as conditions warrant, to give a proper ratio of tube cross-sectional area to header area. By the use of this construction, it is assured that all tubes carry their proper proportion of the total amount of steam passing through the superheater, and the danger of warping or burning of any tubes due to being by-passed is obviated.

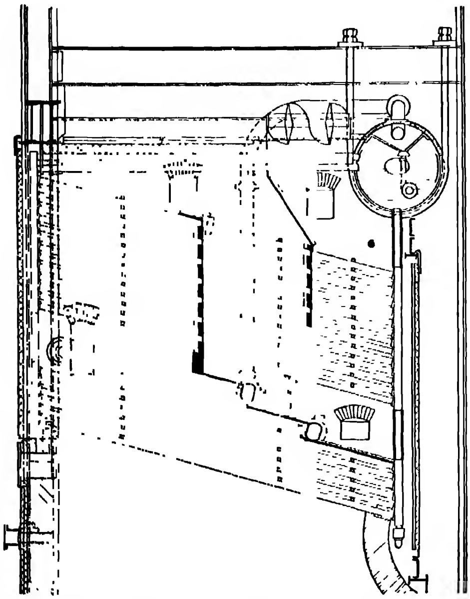

Fig. 86. Babcock and Wilcox Superheater — Located Between Tube Sections.

With single inlet and outlet superheaters, one end of each superheater header is welded closed. To the other end there is attached a wrought-steel flange.

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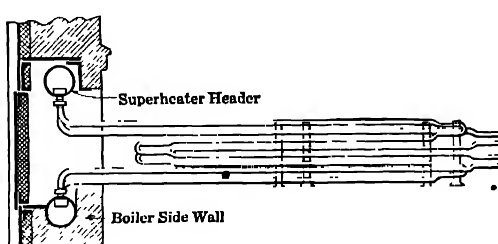

Fig. 87. Assembly of Elesco Superheater — Side Elevation.

ball clamp joint, and no welding, rolling, or gaskets of any kind are applied. The clamp studs are made of special heat-treated alloy steel with a minimum tensile strength of 100,000 lb. per sq. in. Figure 88 shows a section through the header and joints.

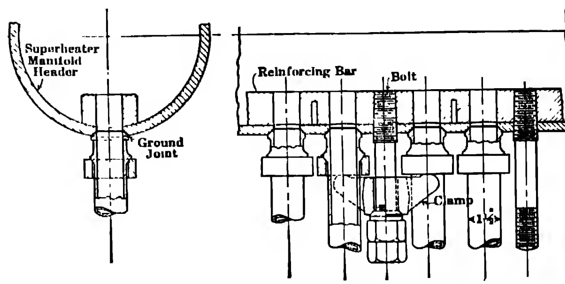

FIG. 88. Method of Securing Tubes to Header — Elesco Superheater.

The elements have forged return bends, an important feature of these superheaters, since it permits placing of elements closer together and better utilizes the space available for the superheater. Owing to the sharp turn, a better mixing of the steam is obtained, resulting in a more uniform heating.

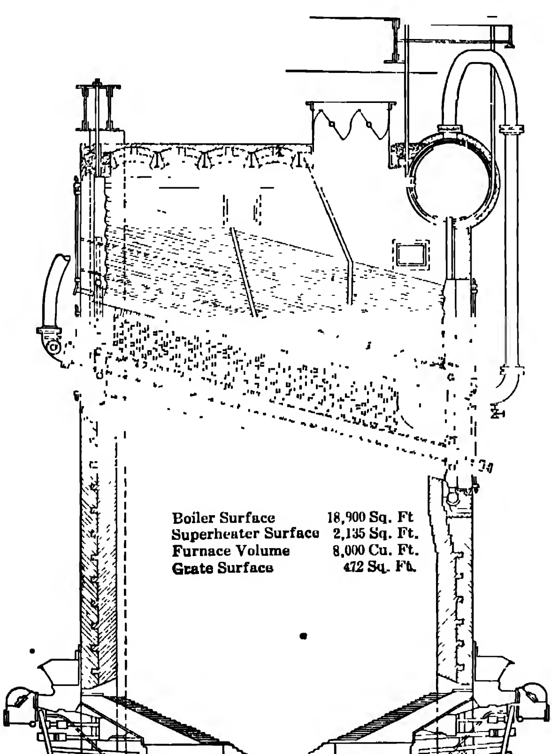

FIG. 89. Elesco Superheater Installation — Hell Gate Station.

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Figures 89 and 90 are examples of two installations in accordance with modern practice in superheater design. Figure 89 shows a large cross-

drum boiler built by the Springfield Boiler Company for the new Hell Gate Power Plant of the United Electric Light & Power Company in New York. The completed plant will consist of 24 boilers, 12 of which are now in service. The superheater is located in the first pass between

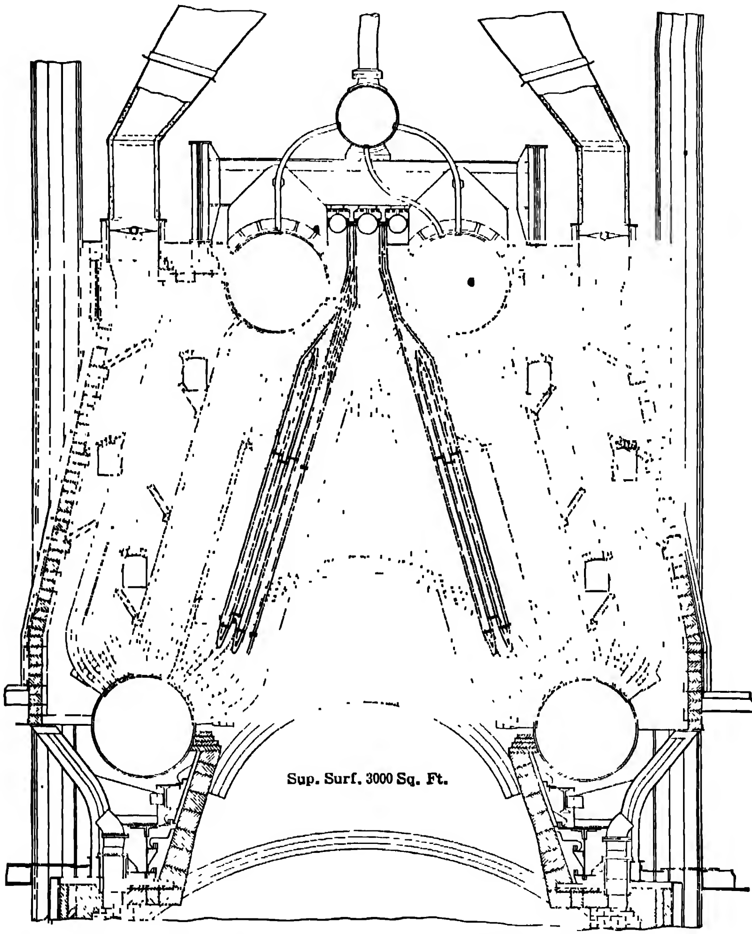


FIG. 90. 26,470 Sq. Ft. Ladd Boiler and "Elesco" Superheater. River Rouge Station.

the sixth and seventh rows of tubes. The space is so selected as to protect the elements from too high temperature and, at the same time, to secure the high superheat specified with a minimum amount of tubing and minimum obstruction to the flow of the gases. The superheater is designed for 200 deg. of superheat at 200 per cent rating. The headers are supported on steel work and are free to expand and contract in all directions. All

joints between headers and units can be inspected from the outside of the boiler by merely removing the access door in front of them.

Figure 90 shows a boiler installed in the River Rouge plant of the Ford Motor Company. Each superheater consists of three headers, two 10-in in diameter and one 12-in. The headers are located below the boiler steam drum. Steam is taken from two points of the boiler, collected, and led to the ends of the 10-in. saturated steam header, both connections being on the same side of the boilers. From the saturated headers, the steam passes through the elements and is returned to the 12-in. superheated steam header and discharged at the side opposite that at which it enters the saturated header, thereby avoiding any possibility of short-circuiting, and at the same time attaining a correct steam distribution through all the elements. The units are placed in the first pass of the boiler, protected by a few rows of boiler tubes. This arrangement not only obtains a location of the superheater in an advantageous gas path, but also practically eliminates the obstruction to the flow of the gases past the superheater. The superheater heating surface is also contained in the least possible space of the boiler. All units are suspended from the headers in a nearly vertical position, and are thus free to expand and contract. The vertical position of the units, coupled with their smooth surface, also prevents the accumulation of soot and ashes. The headers are located outside the boiler away from the hot gases. This location of the headers offers facility for inspection and repairs, even during operation, without the necessity of going into the setting. Any unit in the superheater may be disconnected and removed without interfering with the other units.

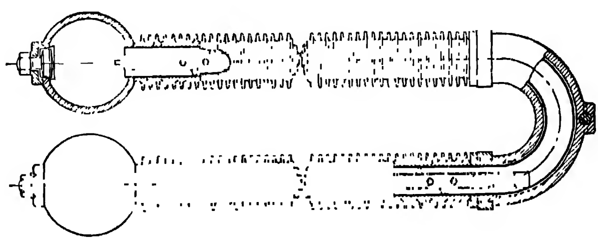


Fig. 91. Assembly of Foster Superheating Element.

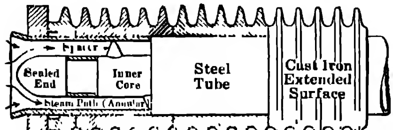


Fig. 92. Details of Construction—Foster Superheater.

Figure 91 shows an assembly of **Foster convection superheater** elements, and Fig. 92 the details of construction. This device consists of wrought-steel headers joined by a bank of straight parallel seamless drawn-steel tubes, each tube being encased in a series of annular flanges placed close to each other and forming an external cast-iron covering of

large surface. The protection afforded by this external covering is ample to prevent damage from overheating during the process of steam raising, and flooding devices are unnecessary. The tubes are double, the inner tube serving to form a thin annular space through which the steam passes as indicated. Caps are provided at the end of each element for inspection and cleaning purposes. Foster superheaters are more costly than plain-tube superheaters, but are longer-lived and offer a much larger heating surface in proportion to the space occupied. Figure 93 shows a vertical section through a **Foster Radiant Heat Superheater.** This superheater is placed so as to form a part of one of the walls or roof of the combustion chamber, absorbing heat, in this position, by radiation from the fire. Each element of this type of superheater consists of a seamless steel tube to the outside of which heavy cast-iron rings are snugly fitted, the individual ring having a flattened side exposed to radiant heat. A steel casing back of the elements forms a solid wall, and the space between the casing and the elements is filled with insulating material. "Shadow" bricks are placed between the elements when it is desired to cut off some of the radiant energy. In the more recent designs, the shadow bricks between adjacent elements have been dispensed with, so that the radiant-heat absorbing surface forms a complete metal wall. The general practice is to install a convection stage superheater of the single loop type, thereby obtaining from 100 to 150 deg. fahr. of superheat in this stage of the superheater. The further rise of temperature is then obtained in the second or radiant stage of the superheater. (See Publication No. 24-74, N.E.L.A.)

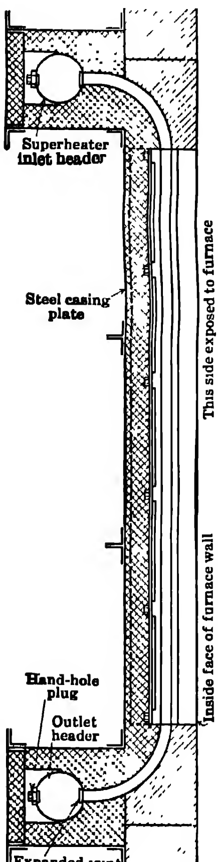


FIG. 93. Foster Radiant Heat Superheater — Side Elevation.

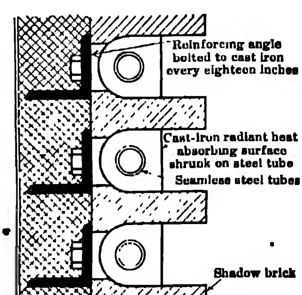


FIG. 93a. Foster Radiant Heat Superheater — Cross Section.

The **Schwoerer superheater,** which is somewhat similar in external appearance to the Foster, differs from it considerably in detail, the heating

surface being made up of suitable lengths of cast-iron pipe ribbed outside circumferentially and inside longitudinally. The ends of the pipes are flanged and connected by cast-iron U-bends. The intention is to provide ample heating surface internally and externally, with a compact apparatus. This superheater design was at one time extensively used in Europe, particularly in South Germany, but is now practically abandoned. It was developed when comparatively low pressures were used, because it was thought that a steel tube was not safe for use with steam in hot temperature zones. It is now, however, universally recognized that the cooling action of steam flowing through a pipe is sufficient to protect the steel tube at gas temperature up to 1500 and 1600 deg. fahr., and no special protection is required.

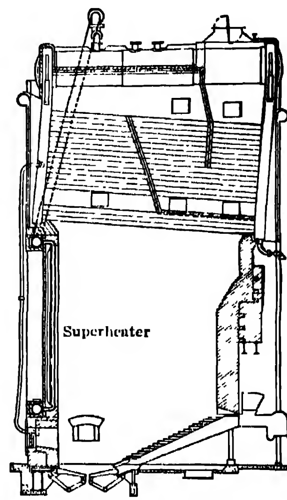


FIG. 94. Installation of Foster Radiant Heat Superheater.

Figure 95 shows the application of a **Heine superheater** to a Heine boiler, illustrating the installation of a superheater within the boiler setting but entirely separated from the main gas passages. The superheater consists essentially of a number of 1 1/2-in. seamless steel tubes, bent to U-shape and expanded into a header box of the same type of construction as

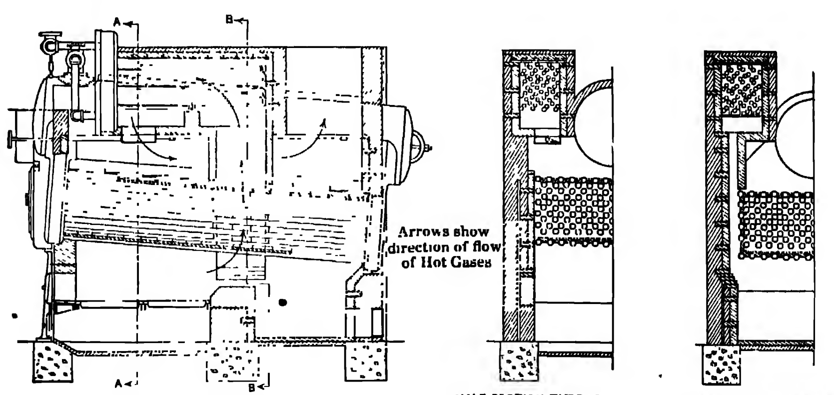


FIG. 95. "Heine" Superheater Installed in Heine Boiler.

the standard Heine boiler water-leg. The interior of this box is divided into three compartments by light sheet-iron diaphragms, so as to deflect the current of steam through the tubes. The superheater

chamber is located above the steam drum as indicated. The gases of combustion are led to the superheater chamber through a small flue built in the side walls of the setting. A damper placed at the outlet of the flue controls the flow of gases and regulates the degrees of superheat. No provision is necessary for flooding the superheating coils since the gases may be entirely diverted from the heating surface. Soot accumulations are readily removed by introducing a soot blower through the hollow stay bolts.

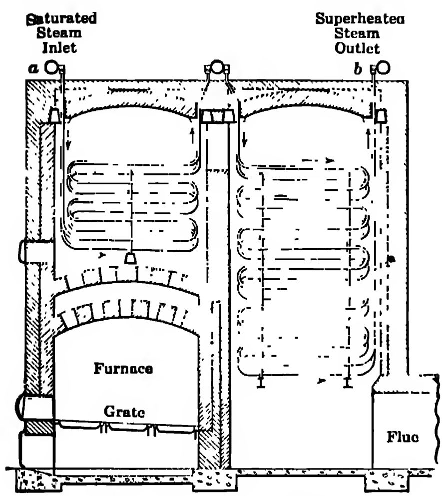


Fig. 96. Elesco Independently Fired Superheater.

A modern, separately fired superheater is illustrated in Fig. 96. The steam enters the superheater at "a" and flows through the lowest pipe of the units, so that the hottest gas comes in contact with the part of the units where the steam temperature is the coolest, and where the steam in most ses still contains some moisture. The units are thus protected against overheating. The steam flow in this part of the superheater is in the same direction as that of the gases, but owing to the great temperature difference between the gases and the steam the heat absorption is still high. In the second section, the steam flows in the opposite direction to the gas, so that the heat absorption here is the highest possible. Where the gases leave the superheater, the temperature of the steam is still comparatively low, and the gases are cooled sufficiently to secure a high efficiency of the superheater. The superheated steam leaves the superheater at "b." This arrangement permits of low flue-gas temperatures and high steam temperatures without subjecting the elements to the severe action of the heat. The design of headers and joints is the same as that of the Elesco Integral Superheater.

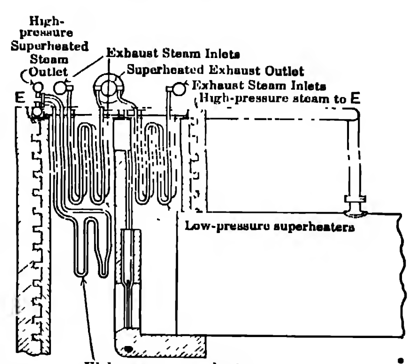


Fig. 97. High- and Low-pressure "Elesco" Superheater.

Figure 97 shows a low- and a high-pressure Elesco superheater located at the rear of a return-tubular boiler. The high-pressure steam, 100-lb. gage pressure, is superheated about 150 deg. fahr. and the exhaust steam, 15-lb. gage pressure, is superheated to 400 deg. fahr. The superheating of exhaust steam for certain industries is productive of marked fuel economy. See "Textile Plant Superheat Exhaust Steam for Process Work," *Power*, June 20, 1922, p. 965.

96. Materials Used in Construction of Superheaters. — In stationary practice, the superheater tubes and headers are ordinarily constructed of mild steel. The tubes are seamless drawn, No. 12 to No. 8 B.w.g. in thickness, and from 1 to 2 in. in diameter. The tubes are bare except in the Foster design. In the latter, they are protected by cast-iron sleeves as shown in Fig. 92. The headers are made of extra-heavy steel pipe or steel boxes of rectangular cross section. In locomotive and marine practice the use of cast steel or high-grade cast iron predominates in header construction. On the continent, cast-iron is used extensively for this purpose.

The effect of temperature on a number of ordinary commercial metals is shown in Fig. 98. It will be seen that the tensile strength drops off very rapidly for temperatures beyond 600 deg. fahr. Because of this rapid decrease in tensile strength of materials with the increase in temperature, high-pressure steam is seldom superheated to temperatures above 850 deg. fahr. Recent development in the manufacture of electric steels has produced cast steel as strong at 1200 deg. fahr. as the ordinary steel at 900 deg. and the yield points also carry the same relation. (*Power*, May 1, 1923, p. 696.)

Properties of Metals at High Temperatures: W. S. Morrison, Trans. A.S.M.E., Vol. 42, 1922.

The Effect of High Temperatures on the Physical Properties of Some Metals and Alloys: The Valve World, Jan., 1913, published by the Crane Company, Chicago.

It was thought that cast-iron valves showed permanent increase in dimensions under high superheat and that in numerous instances they were supposed to have failed altogether, but sufficient data are not available to prove the unreliability of cast iron if the iron mixture is properly compounded and the necessary provision is made for expansion and contraction. As a matter of fact, the behavior of cast iron in connection with high temperatures depends entirely upon the composition of the metal. In cast iron with a considerable amount of silicon, the carbon remains uncombined, as the silicon has a tendency to combine itself with the iron, driving out the carbon. In such cases the carbon is contained in the iron in the form of graphite and the structure of the material is weak.

It may be noted that cast iron is made up of approximately 93 per cent iron and 7 per cent other elements by weight, but iron is so much heavier than carbon, silicon, etc., that by volume it is 69 per cent carbon and 31 per cent of the other materials. The great amount, by volume, of un-

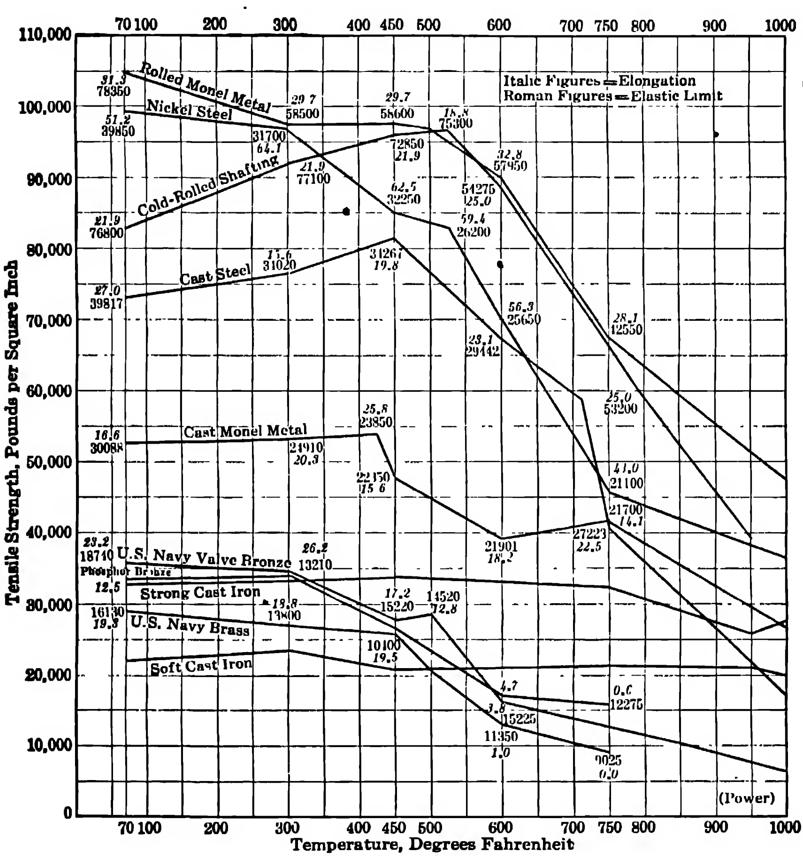


FIG. 98. Effect of Temperature on Strength of Materials.

combined carbon is the cause of the deterioration of cast iron at very high temperatures. The lower the silicon content, the greater the amount of carbon that becomes combined with the iron, the more homogeneous is the metal, and the better it can withstand high temperatures. It is, therefore, now an established fact that the better grade of cast iron can be used for highly superheated steam without any bad effect, and that fittings made of low-silicon cast iron are giving satisfactory service with highly superheated steam. Notwithstanding the claim that cast iron properly

compounded is a perfectly reliable metal for fittings, engineers are inclined to use cast or forged steel, at least for high pressure and temperatures.

97. Extent of Superheating Surface. — The required extent of superheating surface for any proposed installation depends upon: (1) the degree of superheat to be maintained; (2) the velocity of the steam and gases through the superheater; (3) the character of the superheater; (4) the weight of steam to be superheated; (5) the moisture in the wet steam; (6) the weight, composition, and temperature of the gases entering and leaving the superheater; (7) the conductivity of the material; (8) cleanliness of the tubes; (9) design of superheater, or manner in which the gases pass over the heating surface, and location of the superheater.

Since the heat absorbed by the steam in the superheater is equal to that given up by the products of combustion, neglecting radiation, this relationship may be expressed

$$SUd = Wc\,(t_1 - t_2) \tag{54}$$

in which

S = sq. ft. of superheating surface per boiler horsepower (b.hp.).
U = mean coefficient of heat transmission, B.t.u. per sq. ft. per hr. per deg. difference in temperature.
d = mean temperature difference between the steam and heated gases, deg. fahr.
W = weight of gases passing through the superheater per b.hp.-hr.
c = mean specific heat of the gases.
t_1 = temperature of the gases entering superheater, deg. fahr.
t_2 = temperature of the gases leaving superheater, deg. fahr.

Transposing equation (54)

$$S = \frac{Wc\,(t_1 - t_2)}{Ud} \tag{55}$$

The heat transfer from the products of combustion to the steam may also be expressed

$$SUd = wc'\,(t_s - t) \tag{56}$$

in which

w = weight of steam passing through the superheater, lb. per b.hp-hr.
c' = mean specific heat of the superheated steam.
t_s = temperature of the superheated steam, deg. fahr.
t = temperature of the saturated steam, deg. fahr.

S, U, and d as in equation (54).

For wrought-iron or mild steel tubes, U varies as follows:

U = 1 to 3 for superheaters located at the end of the heating surface.
= 4 to 8 for superheaters located between the first and second pass of water-tube boilers.
= 8 to 12 for superheaters located immediately above the furnace in stationary boilers, in the smoke box of locomotive boilers, and in separately fired furnaces.

The above values are only averages for standard vertical-pass boilers as they were built a few years ago. The value of U changes considerably with the velocity of the gases, which means that it changes also with the rating. The present tendency is to design boilers so that the gas velocity increases towards the end of the boiler, and the above figures will be increased correspondingly. Figure 99 shows values of U obtained at different ratings in actual operation with a Stirling type boiler of about 1000 hp. Figure 100 shows the temperature range of the hot gases entering superheaters for various percentages of boiler heating surface passed over before reaching the superheater coils.

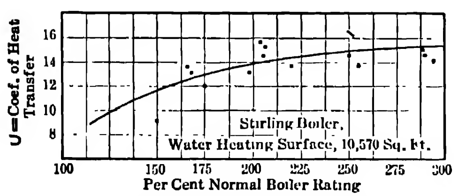


FIG. 99. Coefficient of Heat Transmission — "Elesco" Superheater.

Equations (55) and (56) are only of academic value, since manufacturers of superheaters are more dependent upon experience and judgment than upon mathematical analyses.

In accordance with recent practice, 1 1/2 to 2 sq. ft. of surface per b.hp. is allowed for superheaters located between the first and second passes, and from 3 to 4 sq. ft. for superheaters located at the end of the boiler heating surface, for superheats from 100 to 150 deg. fahr., boiler pressure about 150 lb. The nearer the superheater is located to the furnace the smaller becomes the heating surface, and 3/4 to 1 sq. ft. per b.hp. is not a rare occurrence.

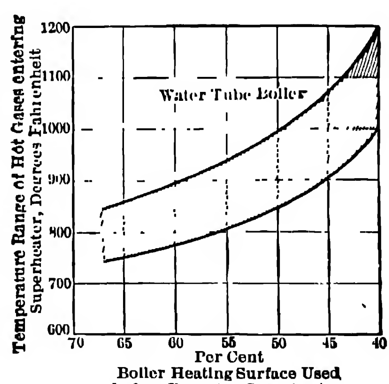


FIG. 100. Temperature Range of Gases in Superheater.

The Power Specialty Company allows 6 B.t.u. per linear foot per degree difference in temperature for their "two-inch" "Foster" element where

the average temperature of the gases is about twice the mean temperature of the steam.

For all engineering purposes, d may be determined with sufficient accuracy from the relationship

$$d = (t_1 + t_2)/2 - (t_s + t)/2 \tag{57}$$

Notations as in equations (55) and (56).

An empirical formula for determining the extent of superheating surface in connection with indirect superheaters, which appears to give satisfactory results for superheaters placed between the first and second passes of vertically baffled water-tube boilers, has been developed by substituting the following values,

$$U = 3, \qquad d = t' - (t_s + t)/2, \qquad w = 30, \qquad c' = 0.5,$$

in equations (56) and (57) (J. E. Bell, *Trans. A.S.M.E.*, 29-267).

Thus:

$$S \times 3\,[t' - (t_s + t)/2] = 30 \times 0.5 \times (t_s - t) \tag{58}$$

from which

$$S = \frac{10\,(t_s - t)}{2\,t' - t_s - t} \tag{59}$$

t', the mean temperature of the product of combustion where the superheater is located, may be approximated from equation:

$$1 \div (t' - t)^{0.16} = 0.172\ H + 0.294 \tag{60}$$

in which

H = the proportional part of boiler-heating surface between the point at which the temperature is t and the furnace.

t as in equation (56).

Equation (60) is based upon the assumption that the heat transferred from the gases to the water is directly proportional to the difference in temperature; that the furnace temperature is 2500 deg. fahr.; flue temperature 500 deg. fahr.; steam pressure 175 lb. per sq. in. gage; 1 b.hp. is equivalent to 10 sq. ft. of water-heating surface; and that there is no heat absorbed by direct radiation. It is, however, present practice to subject a larger amount of the boiler-heating surface to radiation, in order to decrease the furnace temperature with a corresponding decrease of the maintenance of the furnace brickwork. This fact must be taken into consideration in figuring the surface of a superheater.

Example 22. — What extent of heating surface is necessary to superheat saturated steam at 175 lb.-gage pressure, 200 deg. fahr., if the superheater is placed in the boiler setting where the gases have already traversed 40 per cent of the water-heating surface?

Solution. — Substitute $H = 0.4$ and $t = 378$ in equation (60)

$$1 \div (t' - 378)^{0.16} = 0.172 \times 0.4 + 0.294$$

from which

$$t' = 950$$

Substitute $t' = 950$, $t_s = 578$, and $t = 378$ in equation (59)

$$S = \frac{10\ (578 - 378)}{2 \times 950 - 578 - 378} = 2.12 \text{ sq. ft.}$$

Figure 100 gives the probable temperature range of gases entering superheater after passing over a given per cent of boiler-heating surface.

Relation between CO_2 and Superheat: Report of Prime Movers Committee, N.E.L.A., 1923, Part B, p. 251.

98. Performance of Superheaters. — The factors influencing the performance of superheaters are so numerous and so variable that general data for purpose of design are of little value unless all of these factors are given full consideration. If the combined boiler and superheater efficiency were constant irrespective of the boiler capacity, the ratio of gas weight passing over the superheater surface to steam weight passing through the superheater tubes, as well as the ratio of gas to steam velocity, would be constant at all loads. Under such conditions, the superheat would be approximately constant at any boiler rating developed. In the modern high-set boiler, in which the furnace volume, grate surface, and heating surface have been properly co-ordinated, the overall efficiency is approximately constant over a wide range in capacity, and as a result the superheat is also approximately constant. A typical performance curve is shown in Fig. 101. In general, however, as the capacity increases, the weight of steam flowing through the superheater coils increases almost in direct proportion to the capacity, while the weight of gas passing across the superheater surface increases at a rate inversely to the efficiency at various capacities, the weight of gas per lb. of fuel burned remaining con-

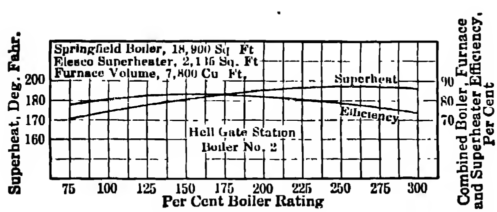


FIG. 101. Performance of Superheater Located Between Tube Decks.

stant. This results in an increased heat transfer rate between the gas and steam. There is also an increase in temperature difference between the gas and steam, so that the combined effect is to increase the superheat as the capacity is increased. This is shown in the curves of Fig. 102.

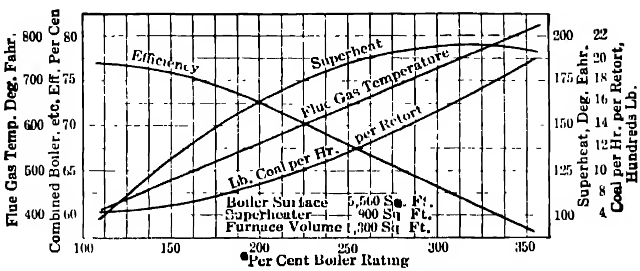


FIG. 102. Performance of Superheater Located at End of First Pass.

This relationship between superheat and capacity will vary widely even with a given fuel and a given set of combustion conditions, with various designs of furnaces, stokers, boilers and superheaters. The variation in superheat is dependent not only on the ratio of the superheater surface to boiler surface, but upon its location with respect to the boiler surface, the action of the flame from the fuel bed, the temperature of combustion, and the kind of fuel. The influence of the kind of fuel on the superheat at various boiler ratings for different fuels and methods of firing is shown in Fig. 103. In Figs. 101 to 103, the superheaters are not exposed to radiation from the fuel bed.

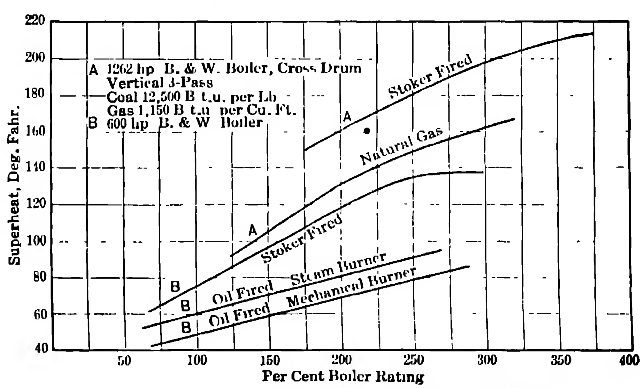


FIG. 103. Influence of Character of Fuel and Method of Firing on Superheat.

Table 31 (taken from *Mechanical Engrg*, Oct. 1921, article by H. B. Oatley) indicates the saving obtained from a superheater installation in a railway power plant. As might be expected, the savings are high because of the extremely poor performance of the engines with saturated steam.

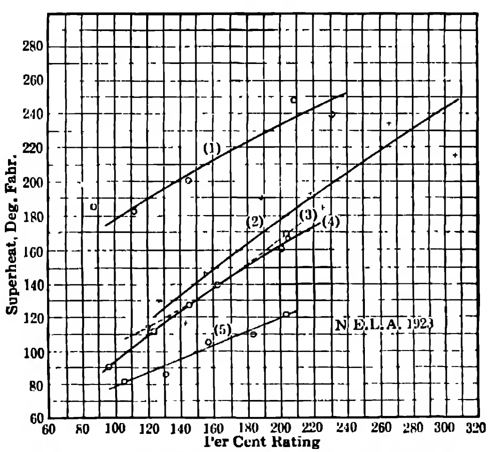
Effect of Superheat on Boiler Performance: Power, Aug. 14, 1923, p. 259.

TABLE 31

TESTS MADE UNDER ACTUAL OPERATING CONDITIONS

Saturated *vs.* Superheated Steam

	Duration of Tests, four hrs. each			
	Saturated Steam	Superheated Steam	Increase Per Cent	Decrease Per Cent
Total coal consumed, lb.	9316	6827	...	26 7
Steam consumption, lb.	57500	46000	.	20 0
Average steam pressure of boilers, lb	118 0	115 0		2.5
Average load on Corliss eng., i.hp.	43 0	43 6	1.4	. .
Average load on Woodmill eng., i.hp	43.2	42 8		0.9
Average load on electric generator, kw	56 7	56.8	0 2	....
Revolutions of air compressor, total	26590	26630	0 1	. .
Boiler horsepower.	386	336	.	12.9
Per cent rated capacity developed	107 0	93 3		...
Temperature of feedwater, deg fahr	212	207		2.4
Heat value of 1 lb. of coal, B.t.u.	13510	13190	.	2.4
Deg. superheat, main header. .		163		...



(1) 6,861 Sq Ft Edgemoor Boiler, 1820 Sq. Ft Foster Superheater, Taylor Stoker, Bituminous & Sub-bituminous fuels.
(2) 11,860 Sq. Ft. B. & W. Boiler, 3,863 Sq Ft. B & W. Superheater, Peabody Mech. Burners.
(3) 6,872 Sq. Ft. Edgemoor Boiler, 1620 Sq. Ft. Foster Superheater, Leahy Burners, Dutch Oven, Front Shot.
(4) Ditto, Flush Front, Rear Shot.
(5) 6,000 Sq. Ft. B. & W. Boiler, 1,277 Sq. Ft. B. & W. Superheater, Hammel Burners, Rear Shot.

FIG. 104. Variation of Superheat with Rating.

Tests were made about two weeks apart and with flues in approximately the same condition.

Some idea of the ratio of superheater surface to boiler surface for a number of well-known central stations is given in Table 32.

TABLE 32

MODERN SUPERHEATER INSTALLATIONS

Station	Type of Boiler	Pressure	Temperature	Boiler Heating Surface, Sq. Ft.	Superheating Surface, Sq. Ft.	Ratio S.H.S. to B.H.S.
Barbados	Springfield	300	625	16,800	1680	0.100
Calumet	B. & W.	1200	750	15,750	2120*	0.134
Do.	B. & W.	350	625	15,089	4052	0.270
Crawford	B. & W.	575	725	11,676	5640	0.339
Cahokia	B. & W.	300	640	18,010	4070	0.225
Colfax	B. & W.	275	650	27,680	5640	0.204
Grand Tower	B. & W.	400	735	11,599	2250	0.194
Kearney	B. & W.	385	680	23,600	4130	0.183
Northeast	Heine	280	650	12,743	5748	0.450
Marysville	Stirling	300	700	28,212	3169	0.112
Hell Gate	Springfield	300	690	18,900	2135	0.113
Philo	B. & W.	650	750	14,086	2427	0.173
River Rouge	Ladd	225	600	26,470	3000	0.113
Springdale	B. & W.	350	700	16,396	2787	0.170
Waukegan	B. & W.	400	700	14,086	2460	0.175
Weymouth	B. & W.	425	700	19,743	2936	0.148
Do.	B. & W.	1200	700	15,730	2923†	0.186

* Primary Superheater, secondary superheater 3300 sq. ft.

† Primary Superheater, secondary superheater 5938 sq. ft.

PROBLEMS

1. A boiler unit generates steam at 250 lb. abs. pressure, superheat 300 deg. fahr. from a feedwater temperature of 210 deg. fahr. What percentage of the fuel burned is required to superheat the steam? Neglect all losses.

2. The average temperature of the products of combustion sweeping past a superheater is reduced from 1000 to 800 deg. fahr. If 12 lb. of gas are produced by each lb. of fuel, how many lb. of steam are superheated from saturation to a final temperature of 600 deg. fahr. for each lb. of fuel burned? Steam pressure 150 lb. gage. Neglect all losses.

3. Required the mean temperature of the products of combustion passing through a superheater of 3815 sq. ft. of heating surface if 66,000 lb. of steam are heated from saturation at 265 lb. abs. to 250 deg. fahr.; mean coefficient of heat transfer, 5; lb. of gas per lb. of steam, 2. Neglect all losses.

4. Approximate the sq. ft. of superheating surface necessary to superheat 10,000 lb. of saturated steam per hr. at 200 lb. abs., to 550 deg. fahr., if the superheater is placed between the first and second pass of a vertically baffled water-tube boiler where the gases have already traversed 35 per cent of the water-heating surface.

5. A 200-hp. boiler, rated at 10 sq. ft. of heating surface per b.hp., generates steam at 135 lb. gage pressure, superheat 100 deg. fahr., feedwater temperature 160 deg. fahr. when operating at 200 per cent rating. If the average temperature of the gases sweeping past the superheater is reduced from 850 to 700 deg. fahr., what is the ratio of superheating to water-heating surface? Use algebraic mean temperature difference and assume $U = 3$.

CHAPTER VI

BOILER SETTINGS, FURNACES, STOKERS AND FUEL BURNING APPLIANCES

99. Settings. — Internally fired boilers and furnaces are generally self-contained and require no separate enclosing or supporting structure other than a suitable foundation. Externally fired boilers, on the other hand, require a **setting** upon which the boiler may rest or within which it may be independently supported by steel framework. The essentials of a boiler setting are a firm foundation to prevent settling and cracking of the walls, proper distribution of masonry and steel work, adequate furnace construction for maintaining efficient combustion and withstanding the stresses due to temperature variation, suitable baffling for obtaining maximum heat absorption and minimum draft losses, and insulation against heat losses. The structural steel work, including metal boiler fronts, inspection doors and frames, and other strengthening and staying devices, are usually furnished with the boiler, but the stoker and masonry construction are generally installed independently, but subject, of course, to the boiler design.

Water-tube boilers are usually uspended from steel work independent of the setting, so that the brickwork supports no load other than itself. Return-tubular boilers under 78 in. in diameter are frequently supported by side brackets or lugs resting directly on the brickwork, but the larger sizes are invariably suspended from steel beams, Fig. 30. The suspended type is by far the better since, by its use, the boiler is free to expand and contract without disturbing the brickwork, and the trouble of brickwork cracking, air leakage and boiler settling is reduced to a minimum. In some of the latest installations, horizontal return-tubular boilers are erected with steel-incased settings — the **barrel,** or **steam-boat,** and the **box** type. In the former the steel plate casing beyond the bridgewall is semi-circular in shape, conforming to the outline of the boiler shell; and in the latter, the outline is that of the standard setting. The steel casings are lined with a layer of fire brick and a layer of common brick, with a thin layer of insulating material next the casing so that heavy brick walls are unnecessary. This type of setting is perfectly air tight and reduces air leakage losses to a minimum.

The side and end walls of a boiler setting should never be less than 12 in.

thick and should be constructed preferably of brickwork, though concrete has been used in some low-pressure installations. Ordinarily the outer walls are built of well-burned red brick, and the inner surfaces, in contact with the hot gases or exposed to the flame, are faced with fire

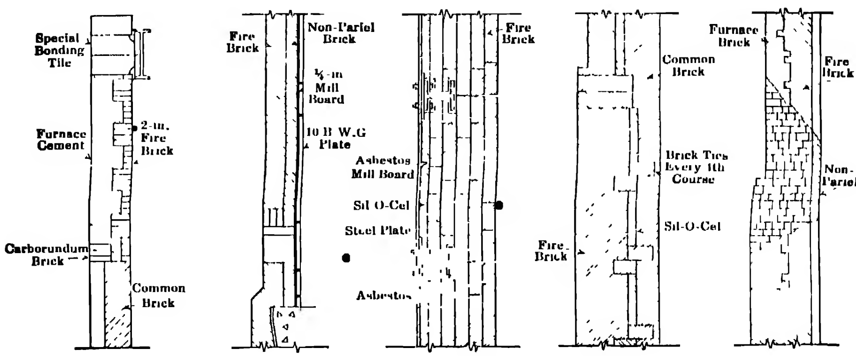


FIG. 105. Examples of Modern Boiler Side-wall Construction.

brick capable of withstanding the high temperatures. In large stoker-fired or oil-fired boilers, the furnace temperatures are very high and solid fire brick or composite walls of various combinations of brickwork, refractory material, heat-insulating coverings and steel jackets are em-

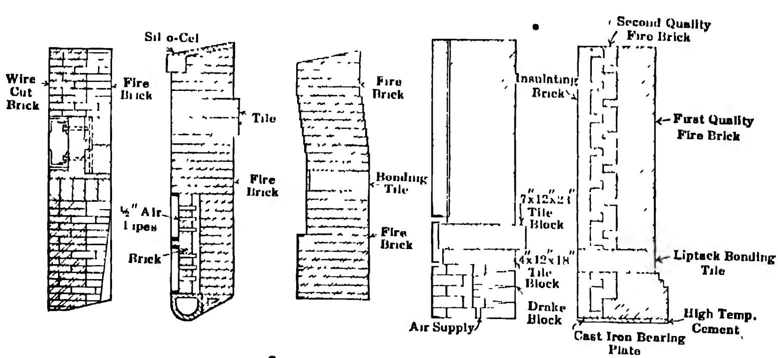


FIG. 106. Examples of Modern Boiler Front-wall Construction.

ployed. There is no general practice, and each installation is designed to meet the particular conditions involved. Some idea of the various combinations for large boiler units may be gained from an inspection of Figs. 105 and 106. Furnace walls, immediately above the fuel bed, are frequently safeguarded and the lining preserved by proper ventilation. This may be effected by the use of special perforated iron blocks, or re-

fractories, or by means of water-circulating pipes or water boxes installed in the side walls immediately above the grate surface.*

The arched construction, forming the roof of certain types of furnaces, is commonly designated as the **furnace arch,** Fig. 129; that immediately over the fire bed, if independent of the roof, the **ignition** or **coking arch,** Fig. 107; and that located beyond the bridgewall, the **deflection arch,** Fig. 116. Ignition arches, as the name implies, are for the purpose of igniting the fuel, and deflecting arches act as mixing devices. In some of the modern high-set boilers, equipped with underfeed stokers, no arches are required, while in certain classes of hand-fired furnaces both ignition and deflection arches are to be found. These arches are either of the **suspended,** Fig. 107, or **sprung,** Fig. 120, type, and are invariably constructed of high-grade refractories in power boilers, and occasionally of water-box construction with refractory lining in low-pressure heating boilers. Ventilation of arches subjected to high temperatures is of great importance; without proper ventilation, the steel supports in the suspended type will become overheated and the refractories in the sprung type will sag and fall.

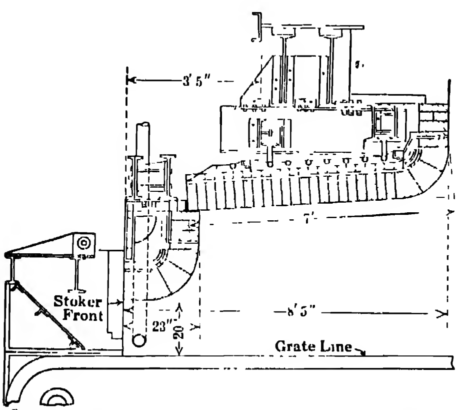


FIG. 107. Suspended Arch - Chain-grate Stoker.

The partitions placed among the tubes of water-tube boilers, for the purpose of directing the flow of the hot gases, are generally known as **baffles.** These baffles may be at right angles to the tubes, Fig. 122 (**vertical baffles**), parallel with the tubes (**horizontal baffles**) Fig. 95, or **inclined,** Fig. 412. They are constructed of cast-iron plates lined with refractory material, specially shaped fire tile or plastic refractory cement. Baffles should be air-tight and yet permit of tube removal, and should be arranged so that the tubes will receive the full scrubbing action of heated gases without short-circuiting. Dead spaces and pockets where soot may accumulate should be avoided, except, of course, where such pockets are installed for the purpose of collecting the refuse. The number and position of the baffles have a marked influence on the boiler efficiency, but there are so many factors bearing directly on what constitutes the proper installation that it is impossible to set any fixed rules. The arrangement of baffling in a number of modern boiler installations will be found in this chapter.

**Water-Cooled Furnaces:* Mech. Engrg., Mar. 1925, p. 197.

100. Furnaces. — The efficient combustion of fuels for steam generation depends chiefly upon the correct design and proper operation of the furnace. For each fuel and set of operating conditions, there is a boiler and furnace equipment which will give the best returns on the investment, but the variables involved are so numerous that each installation must be considered by itself. *Whatever may be the nature of the fuel or the conditions of operation, for complete and efficient combustion, the furnace must be constructed and operated in such a manner that the combustible gases will be brought into intimate contact with the proper amount of air, and maintained at a temperature above the ignition point until oxidation within the combustion zone is complete.*

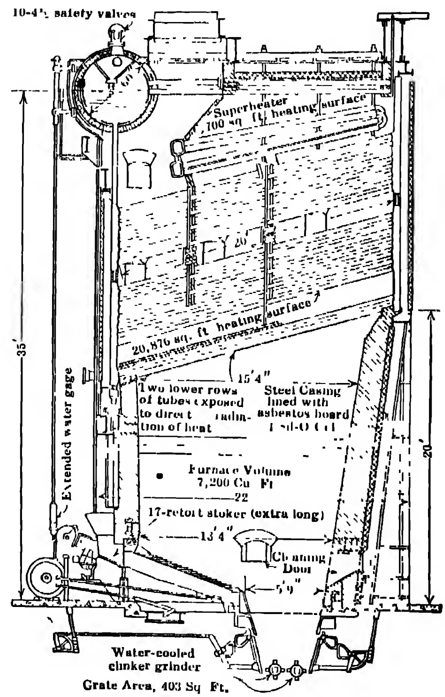


FIG. 108. Boiler and Furnace Equipment — Colfax Station, Duquesne Light Co.

For highest heat efficiency, the temperature of combustion should be the maximum that can be maintained, but the brickwork employed today will fail if subjected to the full temperature available with most fuels. In furnace design, therefore, either some heat efficiency must be sacrificed to maintain the furnace brickwork and keep the cost of repairs within reasonable limits, or the temperature of the walls must be kept below the danger point by exposing a large portion of the boiler-heating surface to direct radiation and by artificially cooling the refractories.

Furnaces for burning oil, gaseous and powdered fuels are of the simplest construction, since the fuel is of such a nature that it can be intimately mixed with the required amount of air and burned in suspension. The dominant factors are furnace volume and length of flame travel.

Furnaces for burning coke, anthracite, and other low-volatile solid fuels require no special provisions for a combustion chamber, except for high boiler ratings, since the fixed carbon of which they are largely composed is burned on or near the grate. Some combustion space, of course, is necessary to provide for mixing the air with the CO rising from the fuel

bed. The greater the volatile content, the larger must be the combustion space, but the increase is not directly proportional to the volatile content.

Bituminous coal and other fuels high in volatile combustible matter require considerable furnace volume, because a large amount of the volatile combustible must be burned above the fuel bed.

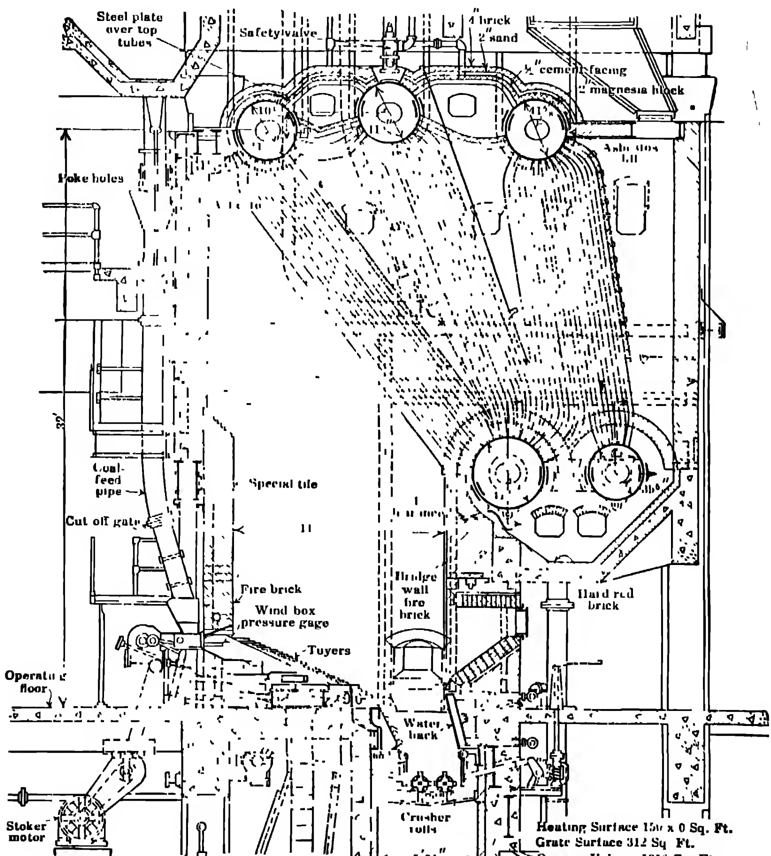


FIG. 109. Boiler Unit No. 4 — Delaware Power Station, Philadelphia Elec. Co.

In burning high-volatile coals, the furnace should be so designed that the distillation of the volatile matter takes place at low temperatures. This favors the formation of light hydrocarbons which are more apt to burn completely without depositing soot than the heavier compounds distilled at high temperatures. Slow and uniform heating of the coal, which causes a large part of the volatile matter to be distilled at low temperature, and *distillation of the volatile matter in presence of oxygen, are the conditions*

productive of smokeless combustion. These conditions are fulfilled by the mechanical stoker. On the other hand, in the hand-fired furnace, distillation usually takes place at high temperatures and in almost entire absence of oxygen, resulting in the production of soot.

When a fresh charge of high-volatile coal is fed into a hand-fired furnace, an enormous volume of volatile matter is evolved. For complete combustion, a corresponding amount of air must be supplied and intimately mixed with the volatile gases before they leave the combustion zone. This requirement for variable air supply makes smokeless and economic burning of soft coal difficult. Furnace volume alone will not give efficient combustion. Complete mixture of air with the combustible gases at high temperature is the all-important factor. In fact, furnace volume is merely a means of effecting good mixture by lengthening the time of contact of gases and air within the proper zone of combustion. An excess of air is required, but the amount can be small if the mixing is thorough.

Furnace volume is defined, by the A.S.M.E. Committee on Power Test Codes for horizontal return-tubular boilers and water-tube boilers, as the cubical contents of the furnace between the grate and the first place of entry into or between the tubes. It therefore includes the volume behind the bridgewall, as in ordinary horizontal return-tubular boiler settings, unless this volume is manifestly ineffective, (i.e., unless there is no gas flow through it) as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is being used. For Scotch, or other internally fired boilers, it is the cubical contents of the furnace, flues, and combustion chamber, up to the place of entry into the tubes.

The **furnace volume,** for maximum commercial efficiency, depends upon the size and character of the fuel, rate of combustion, air excess, direction and length of gas travel, method of admitting air, provision for mixing, and so many other factors that general rules based on only a few of these factors are without purpose.

Furnace volumes per sq. ft. of boiler-heating surface, or per sq. ft. of grate surface, have been increasing very rapidly during the past few years for large stoker installation as well as for pulverized coal and oil fuel and have now reached relatively enormous proportions. Increased furnace volume in externally fired stationary boilers is usually effected by increasing the " head room " or distance from the boiler-room floor to the bottom of the front header. Ten years ago, vertically baffled boilers of the B. & W. type were commonly set with head room as low as 7 ft., while to-day 12 ft. is considered none too high, and 20 ft. has been used in several instances. In the low settings, the ratio of furnace volume to boiler surface is approximately 1 to 10 and in the 20 ft. setting this ratio is nearly 1 to 2. The greater the ratio of furnace volume to boiler surface,

the higher will be the overload capacity and the higher the efficiency at overloads. But high-set units cost a great deal more than those with comparatively low head room, and the first cost and furnace maintenance is much higher. Furthermore, as the volume is increased, first cost and maintenance cost mount steadily, while efficiency increases more and more slowly. The curves in Fig. 110 are of interest in showing the relation

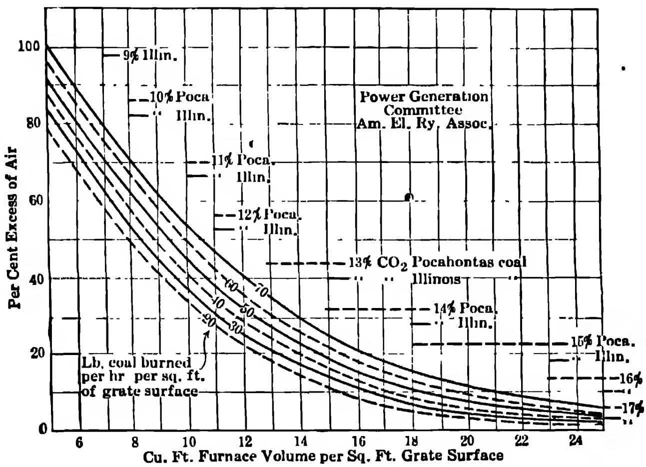


FIG. 110. Furnace Volumes for Illinois and Pocahontas Coals.

between furnace volumes and air excess for Illinois and Pocahontas coals, for the tests described in Bul. 135, U. S. Bureau of Mines. While as based they do not show the exact relation between furnace volumes and rates of combustion for the coals in question for all classes of furnaces, because of the great variation in practice in direction and length of flame travel, etc., they do show that greater furnace volumes are required by high percentages of CO_2 than by high rates of combustion. See also Table 29.

Flow of Heat through Furnace Walls: U. S. Bureau of Mines, Bul. No. 8, 1912.

Boiler Furnace Design: Power, April 17, 1923, p. 613; May 13, 1924, p. 780.

Stoker and Furnace Equipment: Report of Prime Movers Committee, N.E.L.A., Part B, 1923, pp. 126–170; Combustion, Nov. 1924, p. 372.

Factors in the Choice of a Combustion System: Power Plant Engrg., Dec. 1, 1923, p. 1194.

Refractories Service Conditions in Boiler Furnaces: Power, Jan. 19, 1926, p. 113.

101. Grates. — **Stationary grates** for hand-fired furnaces are generally made of cast-iron sections in a variety of shapes, as illustrated in Fig. 111. The bars are ordinarily from 3 to 6 in. deep at the center (this makes them strong enough to carry the load caused by the weight of the fuel without sagging even when the top is red hot), 3/4 in. wide at the top, and taper

to 3/8 in. at the bottom to enable the ashes to drop clear. The width of the air space is determined by the size of the fuel to be used and the air pressure. It is common practice to allow 1/8, 1/4 and 5/16 in. air spaces for No. 3, 2, and 1 buckwheat, respectively; 3/8 in. for pea coal, and 1/2 in. openings for bituminous coal.

The **Tupper** and **Herringbone** grate bars are stiffer and less likely to warp than the common form, but are not so readily sliced, and therefore not so convenient with coal that clinkers badly. Sawdust or pinhole grates are used in burning sawdust, tanbark, and very small sizes of coal. Grates are often set horizontally and the bars are held in place simply by their own weight, but long grates are best placed sloping toward the rear to facilitate firing. The front of the grate, when designed for bituminous coal, is often made solid, this portion being called **dead plate.** It serves to hold the green fuel until the hydrocarbons have been distilled off, when the charge is pushed back on the open grate at the time of the next firing. The length of a single bar or casting should not exceed 3 ft. The length of grate may be made of two or three bars and should not exceed 6 ft. with bituminous coal, as this is the greatest length of fire that can be readily worked by a fireman. With buckwheat anthracite, furnaces 12 ft. in depth are not unusual as anthracite fires require no slicing.

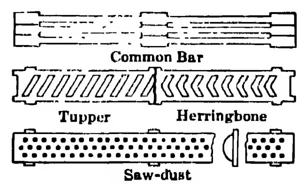


FIG. 111. Types of Grate Bars for Hand-fired Furnaces.

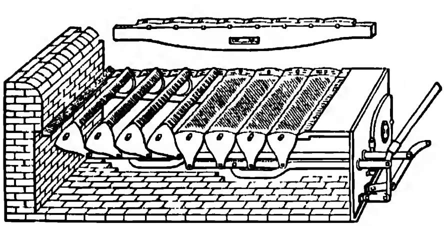


FIG. 112. A Typical Shaking Grate.

The disadvantage of using stationary grates is that the fire is not easily cleaned. Unless the air spaces are kept free of clinkers and ashes, combustion is hindered and the fire rendered sluggish. Frequent cleaning, however, is wasteful of fuel and reduces the furnace efficiency by letting in a large excess of air every time the fire door is opened. In small plants where larger sizes of anthracite are burned, the plain grate is probably as satisfactory as any; the shaking or rocking grate is to be preferred with coals that clinker and have high ash content. Anthracite dust, silt, culm, and screenings are burned on grates with small openings, and require mechanical draft.

Shaking grates have the advantage of permitting stoking without opening the fire door, and require less manual labor than stationary grates. There are many types of sectional shaking grates on the market

and some of them are made self-dumping. A popular type is illustrated in Fig. 112. Each row or section of grate bars is divided into a front and a rear series by twin stub levers and connecting rods. An operating handle is adapted to manipulate either one or both of the levers in such a manner that the front and rear series may operate separately or together. The shaking movement causes no increase in the size of the openings and hence prevents the waste of fine fuel. Ordinarily, the width of the grate is made equal to two or more rows of grate bars, so that the live fire may be shoved sidewise from one row to the other when cleaning. A depth of fire of from 6 to 10 in. is carried, according to the nature of the fuel and the available draft. Manually operated inclined shaking grates, which give the fuel a progressive forward motion, are usually designated as *hand-stokers* (see paragraph 107).

Mechanical Stokers, see paragraph 108-111.

102. Plain Furnace and Hand-fired Setting. — This so-called "**standard**" **setting,** Fig. 113, is intended primarily for anthracite and low-

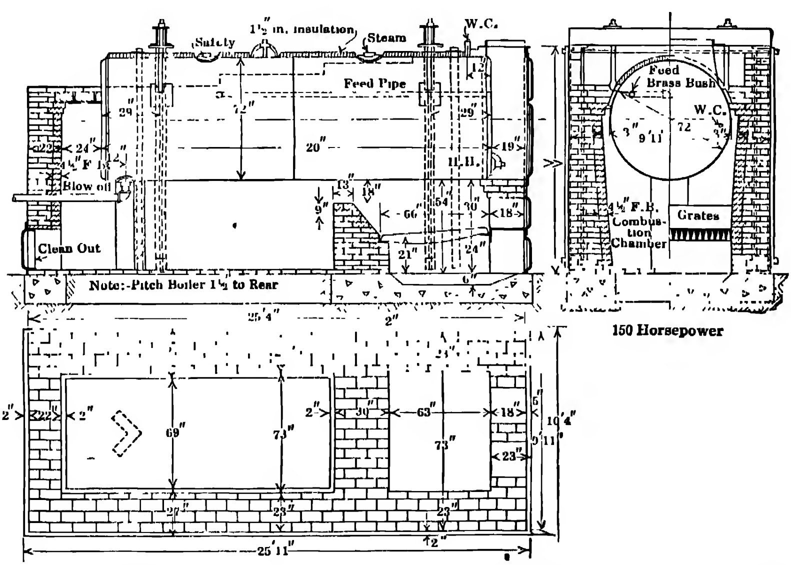


FIG. 113. Return-tubular Boiler with Plain Furnace Setting; Flush Front. (Not Suitable for High-volatile Coals).

volatile bituminous coals. Thousands of these settings are scattered all over the country, and a large number, in the Middle West, are used for burning high-volatile bituminous coal. Anthracite and low-volatile bituminous coals require the simplest type of furnace, because combustion takes place principally in the fuel bed, and, for ordinary rates of driving,

only a small combustion volume is necessary. For this reason the plain setting, which is the simplest and cheapest that could be devised for steam generation, gives satisfactory results with these fuels. With high volatile coal the furnace is inadequate and it is almost impossible to operate the boiler without the production of objectionable smoke, except after the volatile matter has been distilled from the coal. Increased combustion space obtained by raising the boiler may promote better combustion, but the air and combustible gases have a tendency to flow in parallel streams without mixing, and particularly so at low rates of combustion. In ordinary practice, boilers should be set at a height above the dead plate not less than 0.25 of the grate length plus the height of the bridgewall. Head room alone will not effect smokeless combustion, however; mixing devices of some sort are necessary adjuncts.

The installation of this style of setting is not permitted where smoke ordinances are enforced. Fuel economy depends so much upon the

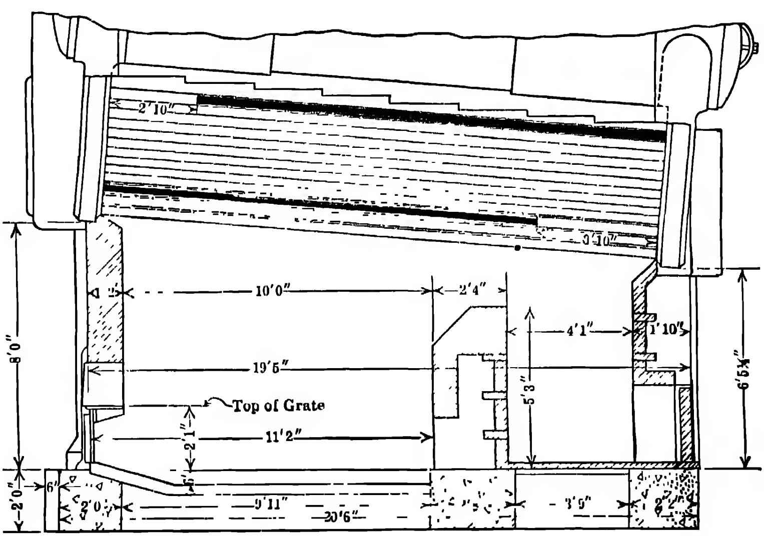


FIG. 114. Heine Boiler with Plain Furnace. Hand-fired. (Modern Setting for Anthracite Coal).

correct design of furnace and setting that, even where there are no restrictions, it is inadvisable to install these "standard" settings without advice from some competent combustion engineer.

A modern design of plain hand-fired furnace, as applied to a Heine longitudinal-drum boiler and suitable for burning anthracite, is shown in Fig. 114.

103. Plain Furnace and Setting with Steam Jets. — The oldest device for reducing smoke in the plain furnace and setting is the **steam jet.** The main purpose of the jet in this connection is to mix the air and gases and insure intimate mixture of the products of combustion. This action is purely mechanical, the steam in itself not being a supporter of combustion. The claims sometimes made that steam increases the calorific power of fuel are, of course, erroneous. When steam comes into contact with incandescent carbon it combines with the carbon, forming CO and CO_2, and the H_2 is liberated. Except in the absence of sufficient air for complete combustion, the CO and H_2 immediately recombine with the oxygen from the air to form CO_2 and H_2O. As the heat liberated by the final combustion of CO and H_2 to CO_2 and H_2O, respectively, is the same as that required to break down the H_2O to CO and H_2, there is no gain in heat. There are conditions, with certain grates of coals and refuse, under which a moderate amount of steam injected through the fuel bed prevents clinkering and promotes complete combustion, but such results are due to increase in available heat and not to increase in calorific power. The heat necessary to superheat the steam to stack temperature must be charged against the coal pile, but the loss may be more than offset by this increase in available heat. There is no question as to the value of properly installed steam jets in maintaining smokeless combustion under certain conditions and with certain classes of fuels, but, as a general rule, they are looked upon as makeshifts by experienced smoke inspectors and others competent to judge them. A plain furnace with steam jet equipment, either manually operated or automatic, will usually average from 8 to 12 per cent smoke density with Illinois coals (see paragraph 354). A smokeless stack is not a true indication of efficient operation, since the air dilution may be excessive and the heat demands of the steam jets may be very great. Since air requirements are greatest at the moment of firing fresh coal, and the demand diminishes as distillation of the volatile matter progresses, steam jets need close regulation for best economy. If permitted to run continuously, as is often the case, they may use considerably more of the energy of the coal than they save by effecting smokeless combustion. Practically all of the so-called "smoke consumers" for hand-fired furnaces depend upon the steam jet, or admission of air only above the fire, for their operation. In most of these the jets are automatic and operate independently of the fireman. The most efficient jets are those based on the injector or siphon principle in which the jet induces a flow of air along with the steam. The steam nozzles are usually placed in the front wall, spaced equally across the setting on 18-in. centers, and are charged downward toward the bridgewall, as illustrated in Fig. 117. Occasionally they are placed in the side wall or even in the bridgewall, but the front

wall construction appears to be the best. Many of the patented smokeless furnaces involving the use of the steam jet do not conform with the requirements of the Chicago Department of Smoke Inspection, chiefly because of faulty furnace design.

Steam jets use from 2 to 15 per cent of the steam generated by the boiler, depending upon the size of the boiler, load carried, number, shape and size of nozzles, initial steam conditions, and whether or not they are permitted to discharge intermittently or continuously. The nozzles should be designed for maximum velocity, since velocity, and not quantity, of steam is the important factor. The weight of steam discharged through the nozzles may be closely approximated by Napier's rule, equation (280). See also Table 51.

104. Hand-fired Dutch Ovens. — One of the earliest attempts at hand-fired smokeless furnace construction for high-volatile coals consisted in placing a full extension **Dutch oven,** Fig. 115, in front of the boiler. This provided a large combustion chamber, but the setting was extravagant in floor space and the intense radiation from the incandescent furnace lining effected a too rapid distillation of the volatile matter from the green fuel. Steam jets placed at the sides of the setting and blowing across the fire assisted in mixing the gaseous products but did not satisfactorily solve the problem. By placing the oven partly (**semi-extension**) or completely (**flush front**) underneath the boiler proper, the extra space requirements were reduced or completely eliminated, but a considerable portion of the heating surface was insulated from the fire at the expense of capacity. The next step was to remove part of the oven roof and expose the boiler surface to the direct action of the fire. This increased the economy and capacity of the setting but still failed to effect the desired result. Plain Dutch ovens for hand-fired service, wherever they may be located, are not productive of smokeless combustion without some sort of stoker or mixing device. Dutch ovens, or their equivalent, are generally used in burning fuels of high moisture content, such as tanbark, bagasse, and wood refuse, in order to provide a large surface of heated brickwork for the distillation of the water.

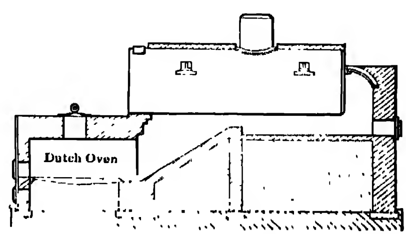


Fig. 115. Plain Hand-fired Dutch Oven —Full Extension. Suitable for Tanbark.

105. Chicago Settings for Hand-fired Return-Tubular Boilers. — Figures 116 to 119 give details of settings for hand-fired return-tubular boilers which conform with the ordinance of the Chicago Department of

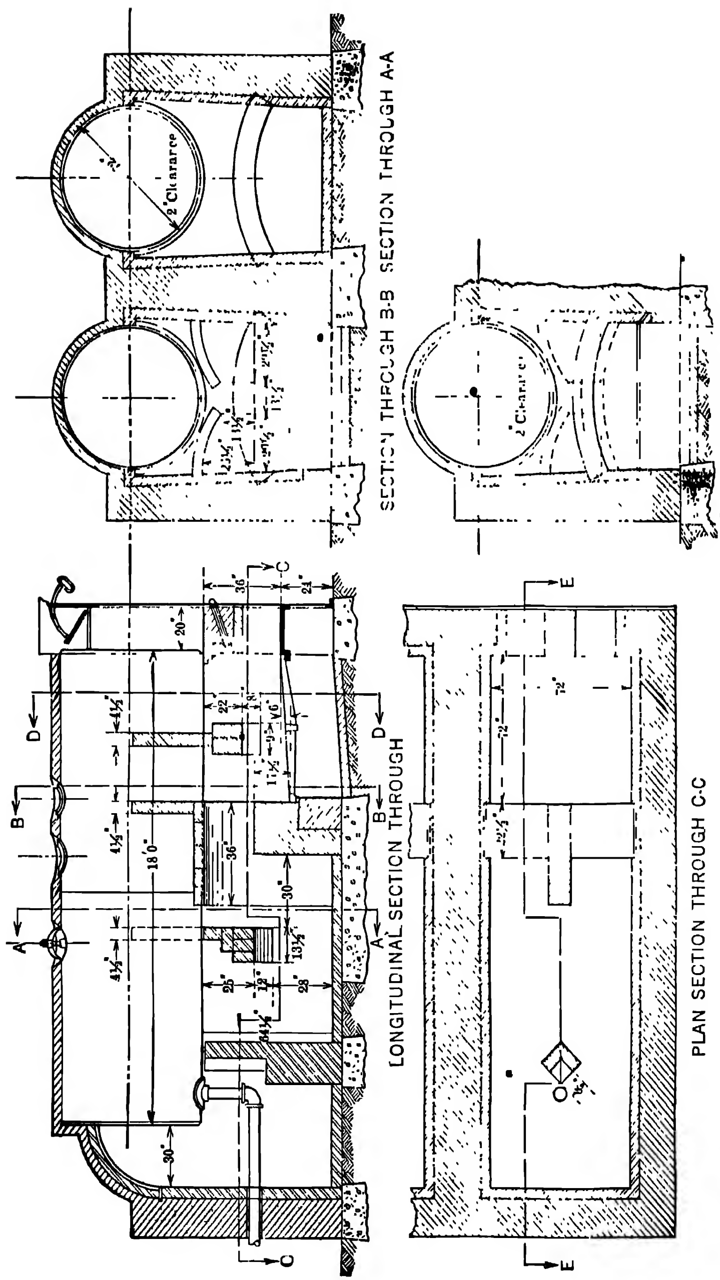


Fig. 116. Double-arch Bridgewall Furnace; Chicago setting.

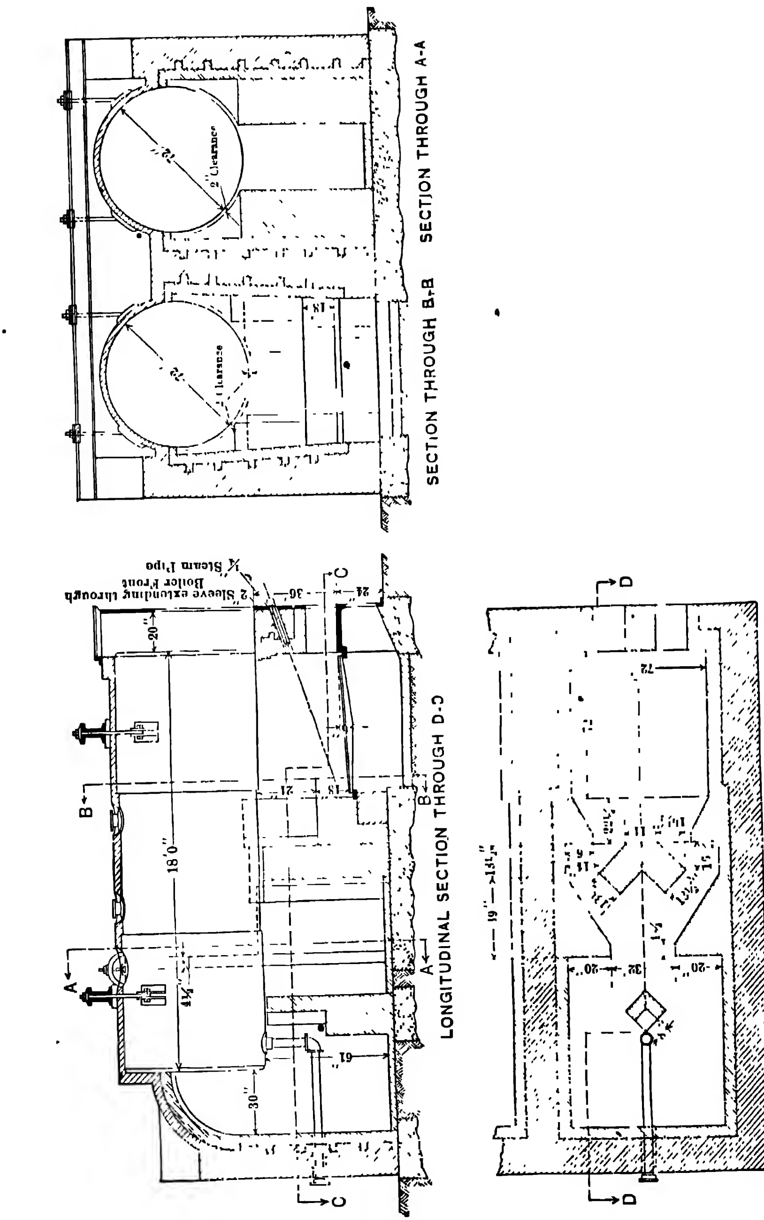


FIG. 117. Department No. 8, or Misostow Furnace; Chicago Setting.

Health, Division of Smoke Inspection. The setting shown in Fig. 116 and known as the **Double-arch Bridgewall Furnace, or Department No. 7 Furnace** (modified), is intended for low-pressure work where steam jets are not effective and where the rate of combustion is 15 lb. of coal per sq. ft. of grate surface per hr. or less. The construction consists of a double arch over the bridgewall (the combined area of the two arches approximating 25 per cent of the grate area) a coking arch over the grate, and a deflecting arch at the rear of the bridgewall. Green fuel is fired in large quantities in front of the coking arch until distillation of the volatile matter is complete. The volatile gases are forced to pass under the coking arch and over the incandescent coke at the back of the grate. From this point the gaseous products are split into two streams by the center pier of the twin arch, and flow in two streams through the two retorts formed by the double-arch bridgewall. On leaving the bridgewall retorts, the gases impinge against the rear or deflecting arch.

The deflecting arch compels the whole volume of gas to change its direction of travel by 90 deg. This arrangement of arches effects an intimate mixture of combustible gases and air at high temperatures, and results in practically smokeless combustion. Auxiliary air is admitted over the fire through panel openings in the fire door. The usual practice is to cut a panel opening in the fire doors having an aggregate area of 4 sq. in. per sq. ft. of grate surface. This type of furnace can be used in connection with horizontally baffled water-tube boilers as well as with horizontal shell boilers.

Figure 117 gives the general dimensions of what is known as the **No. 8, or Misostow furnace,** for high-pressure boilers. Since its adoption approximately 85 per cent of the hand-fired furnaces installed in Chicago have been of this design or modifications of it. The No. 8 furnace consists essentially of a number of vertical fire-brick piers — a center or " V " pier extending from the combustion chamber floor to a point within 2 in. of the shell of the boilers and following the curvature of the shell, and two side piers or wing walls extending from the combustion chamber floor to a point within 2 in. of the shell at the thick part of the pier. The opening between the side walls and the edges of the " V " pier is 25 per cent of the grate area. On the top of the rear end of the wing wall, a 4 1/2-in. bulkhead is constructed for the purpose of forcing the gases to descend and pass between the edges of the two wing walls.

Auxiliary air is admitted over the fire through the fire doors, or in certain cases through the agency of steam jets. The air admitted below and above the grate is forced into intimate contact with the products of combustion by the mixing action of the piers. The dimensions in Fig. 117 refer

to a specific set of conditions and are not general. This furnace should be fired by the alternate method.

Figure 118 illustrates the **Step-down Arch Furnace** as developed by Frank A. Chambers, Deputy Smoke Inspector. As will be seen from the illustration, the furnace construction consists essentially of a series of arches placed in the combustion chamber at a short distance back of the bridgewall. These arches are built in separate rings, independent of one another, and form a series of steps, so that the crown of the last ring is slightly below the top of the bridgewall. The gases, after impinging against the arch, are gradually deflected downward into a high temperature zone formed by the rear face of the bridgewall, the combustion-chamber floor, and the face of the arch. This permits the expansion of the gases, and at the same time effects a very intimate mixture of the

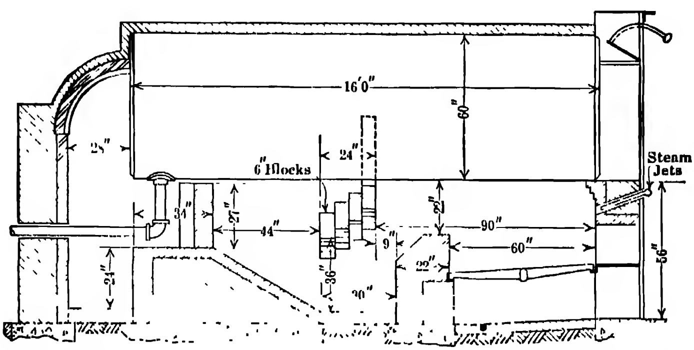


FIG. 118. Chambers Step-down Arch Furnace; Chicago Setting.

products of combustion. The portion of the combustion-chamber floor in the back of the arch is inclined to the rear of the combustion chambers at an angle of 30 deg., and deflects the gases on to the rear end of the boiler shell. It also transmits radiant heat to the front end of the shell. The draft resistance of this construction is very low, and the cost is less than that of any of the other standard furnaces described in this paragraph; moreover, with the exception of the step-down arch, it is a permanent part of the boiler setting, insuring durability and low maintenance cost. This type of furnace is adaptable to both high- and low-pressure boilers; in the operation of the latter, the coking method of firing should be adopted, and in the high-pressure boiler practice, either the coking or alternate method of firing. Steam jets should be supplied as auxiliary equipment where the boiler pressure permits their use, and in all cases, arrangements for air admission over the fire should be made. The dimensions in Fig. 118 are not general and apply only to a specific set of conditions.

The following **head room** requirements, or heights of shell above dead plate, are standard for "Chicago" settings.

Diameter of Shell In.	Dead Plate to Shell In.	Diameter of Shell In	Dead Plate to Shell In.
42	28	66	34
48	30	72	36
54	32	78	38
60	31	84	38

106. Down-draft Furnaces. — Figure 119 shows the application of a **Hawley down-draft furnace** to a Heine water-tube boiler. In this furnace there are two separate grates, one above the other, the upper one being formed of parallel water tubes connected with the water space of the boiler through the steel headers or drums, *A* and *D*, in such a manner as to insure a positive circulation. Fuel is supplied to the upper grate, the lower one, formed of common bars, being fed by the half-consumed fuel falling from the upper grate. Air for combustion enters the upper fire door, which is kept open, and passes first through the bed of green fuel on the upper grate and then over the incandescent fuel on the lower grate. A strong draft is required, owing to the relatively small upper grate area and the correspondingly high rate of combustion. Lump coal gives better results than the smaller sizes, as the latter are apt to fall through the upper grate before being even partially consumed, and when such is the case efficient results cannot be obtained. If carefully manipulated, this furnace, with fire-tiled tubes as illustrated in Fig. 119, gives satisfactory boiler efficiency and smokeless combustion, but its overload capacity is limited. Without the fire tiling, smokeless combustion is possible only at light loads.

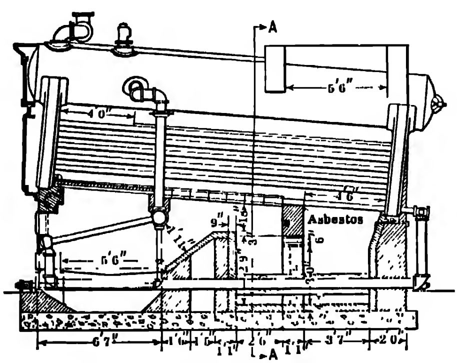


FIG. 119. Hawley Down-draft Furnace.

The down-draft furnace is remarkably successful on low rates of combustion, 10 lb. per sq. ft. per hr. or less, and is used extensively for heating loads. It is not much in evidence in high-pressure plants.

107. Hand Stokers. — Automatic mechanical stoking is unquestionably superior to hand stoking in so far as heat efficiency and smokeless

operation are concerned; but there is a limit in size of boiler below which the overall economy, measured in dollars and cents, may be less with the former than with hand-operated equipment of proper design. Automatic stokers, as a rule, are high in first cost, and if they are applied to small furnaces the fixed charges, maintenance, and operating costs, may offset the saving in fuel. This is frequently the case where the automatic stoker effects no reduction in the firing forces and where the plant operates on a limited hour schedule. There are several types of hand-operated stokers on the market which simulate the action of the automatic mechanical type. When properly installed and manipulated, they are a great improvement over hand stoking, as regards reduction of labor, smokeless combustion and efficiency. The particular equipment illustrated in Fig. 120 consists of a set of stationary inclined grate bars, two rows of **rocking bars** or **pushers,** and two sections of horizontal **dump plates.** The pushers and dump plates are operated from the front of the furnace through the agency of suitable levers.

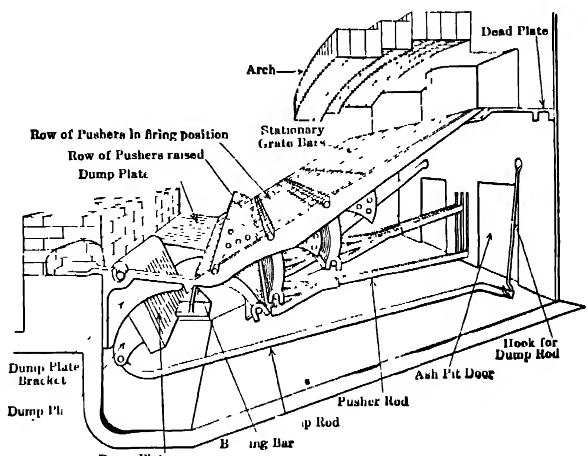


FIG. 120. A Typical Hand-operated Stoker. (National.)

Green fuel is fed to the dead plate and the upper end of the stationary grate, where it is ignited and coked by the heat radiated from the ignition arch. Before a new charge is put in, the coked coal is forced on to the pushers by means of a hoe or shovel. The action of the pushers moves the fuel forward and at the same time breaks up any clinkers. Ash is discharged into the ashpit by lowering the dump plate. Among the well-known types of hand-operated stokers may be mentioned the **Huber, Cokal, National, Auburn, Files** and **Budd.**

108. Mechanical Stokers. — Continuous feeding of the fuel and uniform distillation of the volatile matter in the presence of oxygen are the essential requisites for efficient and smokeless combustion, and it is for this reason that mechanical stokers, as a class, are more effective in producing high combustion efficiency and in preventing smoke than any apparatus accompanied by intermittent firing. In addition to increased

efficiency, they effect a saving in labor and a gain in flexibility of operation. Mechanical stokers, particularly those of the forced-draft type, are capable of responding promptly to high and sudden overloads, and of being brought to full steaming capacity from a banked fire or cold grate in a remarkably short time.

TABLE 33

SETTING HEIGHTS FOR VARIOUS TYPES OF BOILERS EQUIPPED WITH STOKERS

(Min. = absolute minimum, P M = preferred minimum, i.e., the minimum heights recommended.)

(H F. Lawrence)

Type of Boiler	Type of Stoker to be Installed													
	Multiple Retort Underfeed		Single Retort Underfeed				Side Overfeed		Front Overfeed		Chain Grates			
	Taylor, Westinghouse, Riley, Jones A C		Type E		Jones Single-Retort		Murphy Detroit		Roney		Natural Draft		Forced Draft	
	Min.	P.M.	Min.	P.M	Min.	P M	Min.	P M	Min	P.M.	Min.	P.M.	Min.	P.M.
Water-tube.														
Horizontal	10′	12′	10′	12′	8′	10′	8′	11′	8′	10′	10′	12′	12′	14′
Inclined (Hor. M.D.)	7′	8′	6′	8′	6′	8′	5′	7′	6′	8′	6′	8′	7′	8′
Inclined (Vert M D.)	5′	6′	5′	6′	3′6″	5′	3′6″	5′	3′6″	5′	3′6″	5′	6′	8′
Vertical (Hor. M D.)	3′	4′	3′	4′	3′	4	3′	4′	3′	4′	3′	4′	3′	4′
Vertical (Vert. M D.)														
150-hp.	4′6″	5′	4′6″	5′	4′6″	5′	3′3″		3′6″	4′6″	4′1″	4′7″	5′	5′6″
250-hp.	5′6″	6′	5′6″	6′	5′6″	6″	3′3″		3′6″	4′6″	4′1″	4′7″	5′	5′6″
500-hp.	6′	6′6″	6′	6′6″	6′	6′6″	3′3″		3′6″	4′6″	4′1″	4′7″	6′	6′6″
Horizontal Return-Tubular:														
72-in.	8′	10′	8′	10′	7′	10′	7′	8′	6′	8′	7′	8′	8′	10′
84-in.	8′	10′	8′	10′	7′	10′	7′	9′	6′	8′	7′	8′	8′	10′

DEFINITIONS OF SETTING HEIGHTS

Water-tube, horizontal	Floor line to bottom of header above stoker
Water-tube, inclined	Horizontal mud drum: floor line to center of mud drum Vertical mud drum floor line to top of mud drum
Water-tube, vertical	Horizontal mud drum: floor line to center of mud drum Vertical mud drum: floor line to top of mud drum
Horizontal return-tubular	Floor line to under side of shell

Any stoker will burn practically all classes of solid fuels to a certain extent, but no stoker is a *commercial* success with all solid fuels. Some types of stokers are limited to a narrow range in the grade of fuels which

they can burn economically, while others have a wide field of application. For each fuel and set of operating conditions, there is a stoker and furnace equipment which will give the best commercial return on the investment; but the problem of selection is not always a simple one, as is evidenced by the number of changes made from time to time in the furnace equipment of some of our most modern installations. The following outline gives a classification of a number of well-known American mechanical stokers:

TRAVELING OR CHAIN GRATE

Natural Draft		*Forced Draft*	
Babcock & Wilcox	Laclede-Christy	Coxe	Harrington
Green	McKenzie	Illinois	Babcock & Wilcox
Illinois	Playford	Stowe	Westinghouse

OVERFEED

Frontfeed		*Sidefeed*	
Roney	Wilkinson	Murphy	Model
Wetzel	Harrington "King Coal"	Detroit	

UNDERFEED

Single Retort		*Multiple Retort*	
Jones	Roach	Riley	Westinghouse
Type "E"	Moloch	Taylor	Detroit
		Jones, A. C.	Moloch

Sprinkler
Dayton

Any arrangement of the various groups of stokers with reference to their commercial adaptability to the burning of the different classes of bulk fuels is unsatisfactory, because of the great variation in the size, moisture and ash content, and composition of any particular class of fuel. The mere ability to burn fuel is not an index of the commercial success of a stoker equipment, since such items as first cost, maintenance, disposition of ash and clinker, capacity, fuel burned in banking, and ability to meet sudden changes in load, must be given proper weight.

The natural-draft chain-grate stoker is highly successful in burning all bituminous coals which do not require agitation, such as the middle western coals. In fact, agitation during ignition frequently results in the formation of objectionable clinker, particularly with coals having low-fusion-point ash. Eastern and other coking coals may also be satisfactorily burned with natural-draft chain-grates provided with agitating plates, or with forced-draft chain-grates where a high temperature can be maintained at the front of the furnace where the fuel enters. These coals, however,

are better adapted to the overfeed and underfeed stokers which provide a sufficient agitation to keep the fuel bed broken up in a uniform and porous condition. River coal, small sizes of anthracite, culm, coke breeze, bone coal, and low-grade bituminous coals have been satisfactorily burned with forced-draft chain-grate stokers, and many installations of natural-draft stokers are giving excellent results in burning lignites and the high-moisture-and-ash coals of Iowa, Colorado, Montana, Wyoming, Alberta, and Saskatchewan.

Properly installed overfeed stokers of either the frontfeed or sidefeed type are adaptable to almost every variety of bituminous fuel and have been used successfully with lignite and various other fuels mixed with coal, such as tanbark, wood refuse and coke breeze. Coals of low ash content do not produce an ash layer of sufficient thickness to protect the grate bars, and careful manipulation is necessary to prevent the metal from burning. Ignition arches are necessary with all natural-draft overfeeds; and, at high rates of combustion, the fuel is apt to avalanche and considerable annoyance is experienced with clinker because of the high temperatures under the arch. Overfeed stokers are not much in evidence in the large modern central station.

All underfeed stokers are well adapted for burning high-grade caking and low-ash free-burning coals, and the great majority of the modern eastern power plants are equipped with stokers of this type. Underfeed stokers of the self-cleaning type are used to a limited extent with the high-ash free-burning coals of the Middle West but are not as satisfactory as the chain-grate. With proper furnace construction, underfeed stokers may burn small sizes of anthracite or culm when mixed with a certain percentage of bituminous and lignite, but the forced-draft chain-grate appears to be the better investment for these fuels.

Setting heights for various types of boilers equipped with stokers, as specified by H. F. Lawrence,[1] are given in Table 33.

Stoker Equipment and Furnaces: Report of Prime Movers Committee, N.E.L.A., 1923 (Part B), p. 126.

109. Traveling or Chain-grate Stokers. — The chain-grate stoker is one of the most popular forms of automatic stokers for burning small sizes of free-burning coal from the Central States, and is highly successful in burning lignites and many classes of low-grade coals which do not require agitation during the distillation process. While differing in details of construction and in method of driving, the various types of chain-grate stokers, natural or forced draft, are basically identical in general design. The stoker proper consists essentially of a wheel-mounted truck equipped

[1] *The Design and Operation of Underfeed Stokers:* Trans. A.S.M.E., Vol. 44, 1922.

with an endless chain of grate bars. The chain is carried over sprockets at the front and rear ends of the truck and is guided and supported by suitable guide rails or slides. In the older designs, the driving mechanism consists of a gear train actuated by ratchet and pawls, the arm carrying the latter being given a reciprocating motion by an eccentric mounted on a line shaft. The line shaft may be driven by any type of engine or motor, and the speed of the grate (1 to 12 in. per min.) regulated by varying the stroke of the arm carrying the pawls, or by varying the speed of the driving motor. In the new designs, the line shaft and eccentric are dispensed with, and the driving motor is geared to the grate. In the latter case, a variable-speed motor, or a constant-speed motor actuating a variable-speed transmission device, is necessary. The power required to drive traveling grates is very small and ranges from 1 to 15 hp. depending upon the size of stoker and rate of feeding. In the majority of the older **natural-draft** designs, air flows through the entire upper and lower chain as in any stationary grate, while in the newer types, the flow is regulated by a series of independently controlled dampers placed immediately below and traversing the rear half of the upper chain. In all **forced-draft** types, air is forced through the upper chain only, the flow being distributed by a number of separate compartments, each under damper control, occupying the entire space between the chains. These compartments communicate on one or both sides to a common air duct. In some designs any compartment can be operated on forced or natural draft or closed off entirely. In all chain-grate stokers the resistance of the fuel bed decreases toward the rear end of the grate. With the multi-compartment forced-draft type, the air pressure can be regulated to meet the variation in resistance and thereby effect proper combustion with minimum air

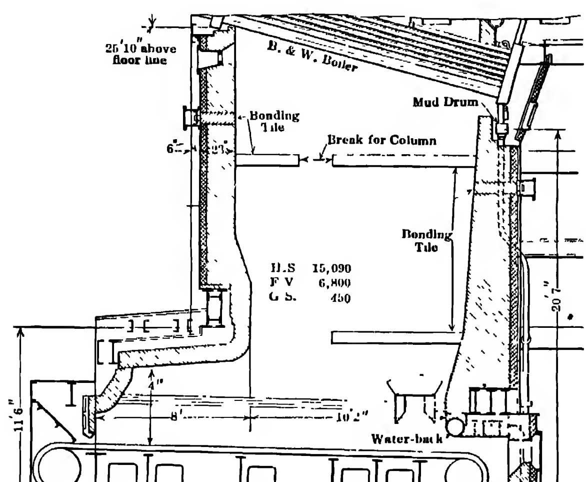


FIG. 121. Coxe Stoker Installation. Calumet Station, Commonwealth Edison Co.

excess. As pressures higher than 2-in. of water are seldom necessary, the power requirements for forced draft are less than with an underfeed stoker of the same capacity. Air leakage around the sides and back end of the chain and air excess through the thinner fuel bed at the rear of the grate is guarded against in several ways. Leakage around the sides is reduced by **adjustable ledge plates** imbedded in the side walls and making

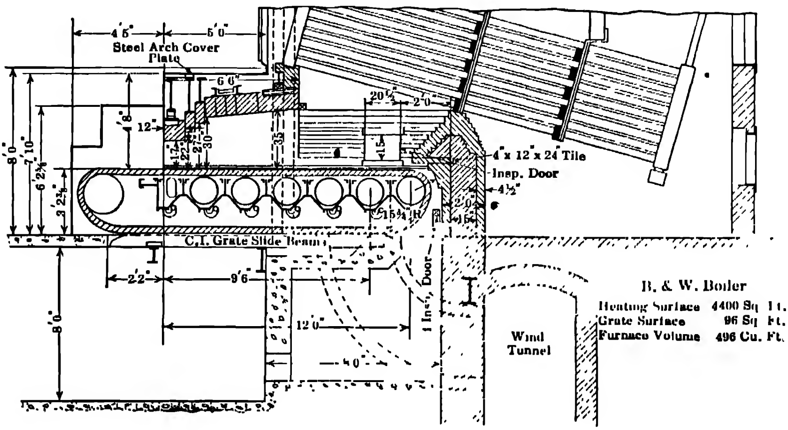


FIG. 122. Illinois Forced-draft Chain-grate Installation for Burning No. 3 Buckwheat Anthracite.

a rubbing seal with the stoker chain. In some forced-draft installations, water boxes are imbedded in the side walls immediately above the grate, so as to prevent the formation of clinkers. Air excess through the rear end of the fuel bed is reduced in some natural-draft designs by the insertion of sheet-metal dampers or baffles below the upper chain, and in others by a **water back** which compresses the fuel bed, making the rear portion denser than the front. In the forced-draft type, the supply of air to each compartment may be controlled to meet the corresponding resistance of the fuel bed, and in case of a short fire the draft may be cut off entirely. Leakage around the end of the chain is prevented by a water back or by swinging dampers and stationary baffles or seals. The water back also presents a water-cooled surface to which clinker will not adhere, eliminates burning off the bridgewall, retains the incandescent carbon on the grate until it is more thoroughly burned, and decreases furnace maintenance. Water backs usually form part of the boiler-heating surface, since the heat absorbed by the water in passing through the box is from 1 1/2 to 5 per cent of that absorbed by the entire boiler, but in some cases the cooling-water supply is independent of the boiler.

All chain-grate stokers require an **ignition arch** for the double purpose

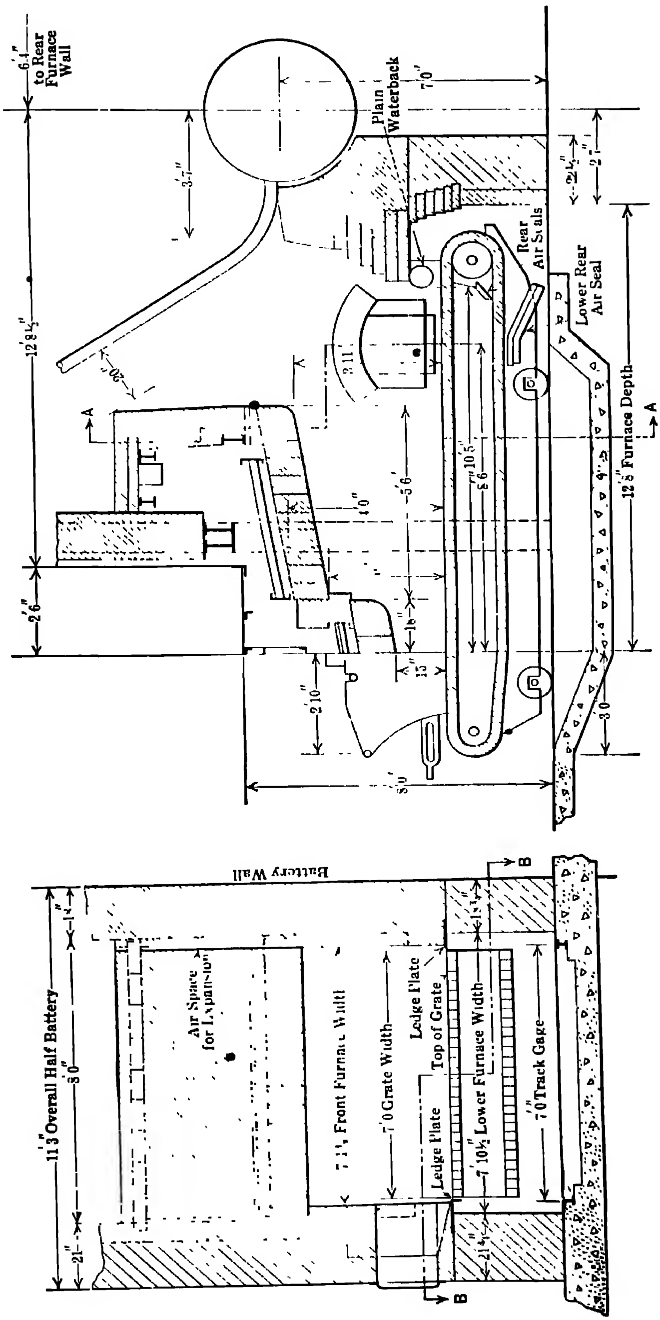


FIG. 123. Green Natural-draft Chain-grate Installation.

of igniting the incoming fuel and directing the products of combustion into the lower portion of the heating surface of the boiler. The width of the arch is dependent upon that of the grate, but the weight, length, and slope are functions of the desired rate of combustion, percentage of volatile combustible in the fuel, calorific value of the fuel, and the stoker length. While general rules are available for approximating the correct proportion of ignition arches, they should be considered only for preliminary layouts because of the great number of variables not included in these rules. Stoker manufacturers are in a position to fur-

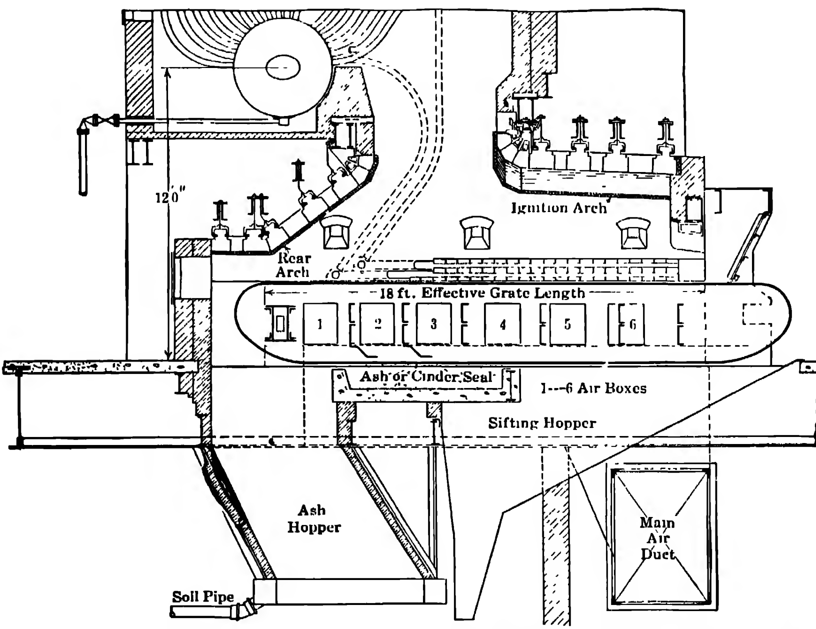


FIG. 124. Harrington Forced-draft Traveling-grate Installation for Burning Coke Breeze. Longitudinal Section.

nish specific data and they should be consulted before adopting any final design. The curves in Fig. 126, compiled by T. A. Marsh, give some idea of the relation between the length of arch and ignition rate for free-burning bituminous coal containing 25 per cent volatile matter. Dimensions of arches and furnaces for burning a few classes of fuels with chain-grates are given in Fig. 121–125. It will be noted that the length of stoker as usually installed is such as to require a furnace extension beyond the front line of the boiler wall.

The chain-grate burns coal progressively and the operation is entirely automatic. The green fuel enters the furnace at one end, passes through

the various stages of combustion, and ash is discharged from the furnace at the other end. Since the fuel and chain move together there is no agitation of the fuel bed, an ideal condition for free-burning fuels. Caking coals, however, usually require agitation during the ignition stage because of the swelling and fusing action of the fuel under the ignition arch, and for this reason the natural-draft chain-grate is not suitable unless provided with coking plates immediately under the front of the arch. Combustion rates up to 40 lb. per sq. ft. per hr. can be secured with this arrangement, but, above this rate, the ashpit loss increases rapidly, and burning of the grate surface becomes serious.

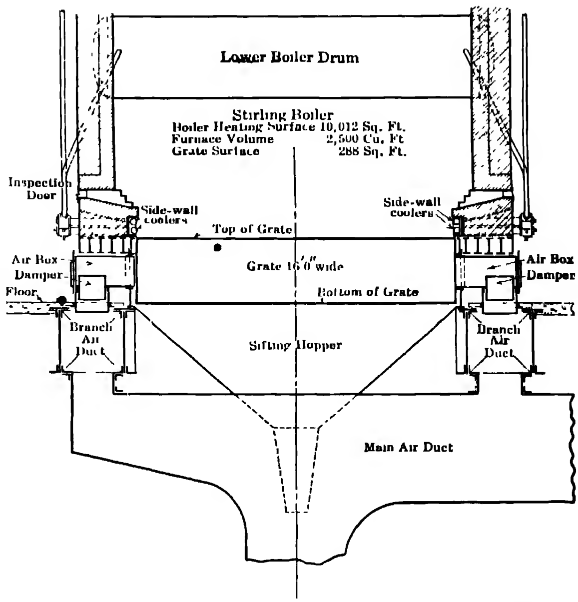


FIG. 125. Harrington Forced-draft Traveling Grate. Side Sectional Elevation.

Forced-draft chain-grates burn caking coals satisfactorily without agitation, because the fuel bed can be increased to the proper thickness and the air supply regulated so as to maintain a high temperature where the fuel enters the furnace. This permits combustion of the volatile matter without caking of the solid particles.

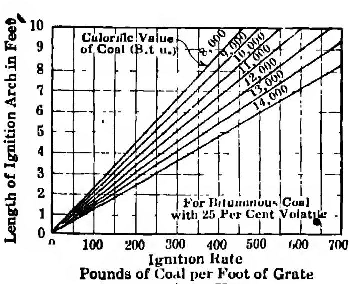


Fig. 126. Relation of Arch Length to Ignition Rate. Natural-draft Chain-grates.

Natural-draft chain-grates are generally installed where the capacities demanded to meet the station load are within range of the natural draft available and where the load demand is steady, or where peaks can be anticipated sufficiently far ahead to permit building up furnace conditions to meet them. Rates of combustion of various fuels with chain-grate stokers are given in Table 26. Several types of stokers with furnaces for burning

different classes of fuels are shown in general detail in Fig. 122 and Fig. 125. See also paragraph 78.

The Development and Use of the Modern Chain Grate: T. A. Marsh, Trans. A.S.M.E., Vol. 44, 1922, p. 773.

Burning Slack Containing Excess Moisture: Power, Apr. 2, 1918, p. 472.

Peak Loads on Chain Grate Stokers: Power, July 1, 1919, p. 20.

Burning Lignite on Forced-draft Chain Grates: Power, Dec. 16, 1919, p. 798.

Stoker Equipment and Furnaces: Report of Prime Movers Committee, N.E.L.A., Part B, 1923, p. 127–137.

Burning Sawmill Refuse on Forced-draft Chain Grates: Power, Oct. 16, 1923, p. 616.

Improving a Chain Grate Boiler Furnace: Power Plant Engrg, Mar. 1, 1924, p. 272.

110. Overfeed Stokers. — In stokers of the overfeed type, coal is pushed in automatically at the top of a sloping grate, coked by the aid of an ignition arch, and fed downward progressively by the movement of the grate bars aided by gravity. Ash and clinker collect at the bottom, where they are crushed by rolls or dumped. Overfeed stokers are used with all grades of coal and are quite common in the older central stations and in power plants where the boiler units are not very large. Arches of the sprung or suspended type are used for ignition and coking. There are two basic types of overfeed stokers: the **frontfeed,** in which the grates slope from the front to the rear of the furnace; and the **sidefeed,** in which the grates are inclined from the side to the center of the furnace, forming a V-shaped receptacle.

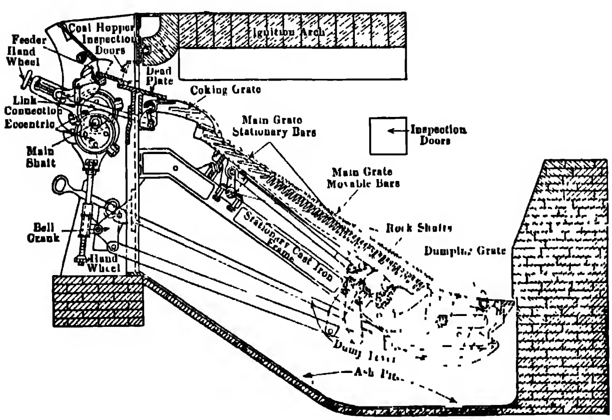


FIG. 127. Wetzel Overfeed Stoker — Sectional Elevation.

The **Wetzel stoker,** Fig. 127, operates on natural draft and is of the frontfeed class. It consists of a cast-iron front on the outside of which is arranged the coal hopper, driving mechanism, and regulators; on the inside, a frame upon which are assembled the coking grate, main grate and dumping grate. Every alternate grate bar of the main grate is movable, and the intermediate bars are stationary. Coal is fed into the hopper, from which it is automatically pushed to the dead plate and coking grates,

where ignition and distillation take place. The coking grate, driven by a link connection to the pusher, moves the coal on to the main grate, where the sawing action of the movable bars causes it to travel slowly down the incline to the dumping grate. When sufficient ash has been accumulated on the dumping grate, a lever is thrown from the front and the ash is discharged into the ashpit. The coking grate contains a very large percentage of air space, the upper part of the main grate a somewhat smaller percentage, the lower part still less, and the dumping grate a very small percentage. Other well-known makes of the frontfeed type are the Roney and the Wilkinson.

In the **Roney** stoker, the grate bars are placed horizontally to the frame and form a series of steps. Each step is rocked back and forth between a horizontal position and an inclination toward the back of the furnace, thus pushing the burning fuel downward from step to step.

In the **Wilkinson** stoker, the inclined grate bars are hollow and are arranged side by side, every alternate bar being movable. When in motion there is a constant sawing action of the grate bars. A small steam jet is introduced into the end of each hollow grate bar and induces a large part of the air required for combustion. This stoker is intended primarily for the burning of fine anthracite coal.

Most of the overfeed stokers are of the natural-draft type and are seldom operated at more than 200 per cent of rated boiler capacity. The **Cleveland** and **Reagan** overfeed stokers are exceptions and are intended primarily for forced draft. Boiler ratings of 350 per cent have been realized with the forced-draft type when burning a good grade of bituminous coal.

Figure 128 shows longitudinal and vertical sections through a **Harrington "King Coal"** automatic stoker as applied to a 72-in. by 18-ft. horizontal return-tubular boiler. It may be operated either as a forced-blast or natural-draft stoker, and is designed for bituminous or lignite coals.

There are four steps in the grate surface, arranged as follows: The first step is the feed plate with attached grate bars, which forms the floor of the hopper and which serves to feed the fuel on to the active grate surface; the second step is stationary; the third step reciprocates like the first, and the fourth step also is stationary. The various steps are formed of grate bars easily removed, having 10 per cent of air space. Being designed for heating service, this stoker is independent of high-pressure steam. It is driven by an hydraulic or electric motor, and the forced blast, when required, is provided by a motor-driven fan. The fuel travels from the hopper toward the rear over the successive steps, the ash being discharged from the rear, or fourth step, on to the ash extractor. This is a plate which reciprocates adjustably and causes the

ash to work forward and finally fall over the front end into the pit. The speed of this plate is so adjusted as to keep the throat full of ash at all times, thus automatically sealing the ash exit against the admission of air. This device is built in sizes from 4 to 40 sq. ft. and costs but little more than a high-grade hand-operated stoker.

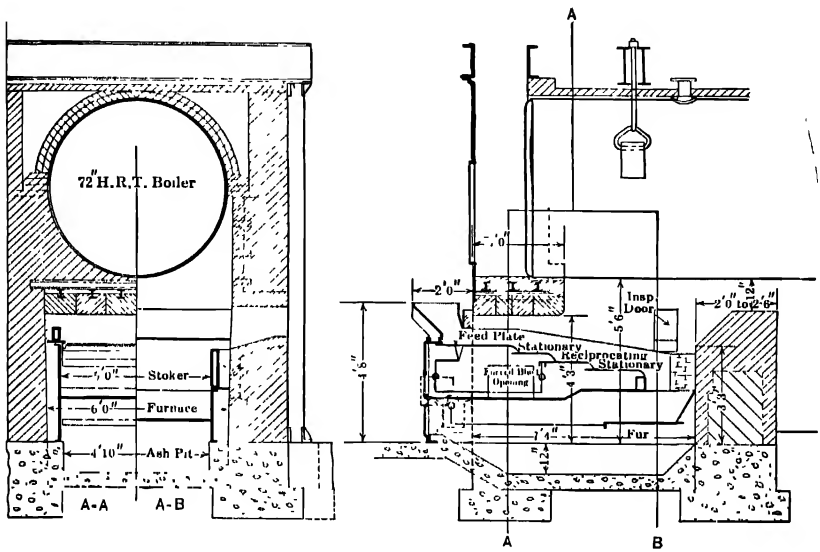


FIG. 128. Harrington "King Coal" Stoker.

The **Murphy, Model,** and **Detroit** stokers are of the sidefeed type and operate with natural draft. The Murphy stoker, Fig. 129, is in effect a Dutch oven equipped with an automatic feeding and stoking device. Coal is introduced either mechanically or by hand into the magazine at each side of the furnace and above the grate and descends by gravity upon the coking plate. Reciprocating stoker boxes push the coal upon the grate bars. Every alternate grate bar is movable and pivoted at its upper end. A rocker bar, driven by a small motor or engine, causes the lower ends to move up and down, this action producing the required stoking effect. A device for grinding up the clinker and ash is provided as shown at the bottom of the furnace. Preheated air is supplied to the green coal through air ducts in the arch plate, and the speed of the stoker boxes and grate bars can be regulated to conform to any rate of combustion. Stokers of the sidefeed type are characterized by large stoking space per foot of grate area and an ample combustion chamber. Under careful operation, they operate smokelessly with free-burning coals from the Middle West, up to 200 per cent of rated boiler capacity. Because

of the high furnace temperature, considerable manipulation by the firemen is frequently necessary in clearing the grate of clinker.

Overfeed Stokers of the Inclined Type: Trans. A.S.M.E., Vol. 44, 1922, p. 787.

111. Underfeed Stokers. — Underfeed stokers utilize the gas-producer principle. Green coal is fed to the lower layer of the fuel bed and is gradually pushed up and coked, giving up its volatile constituents and becoming incandescent by the time it reaches the top layer. The ash or

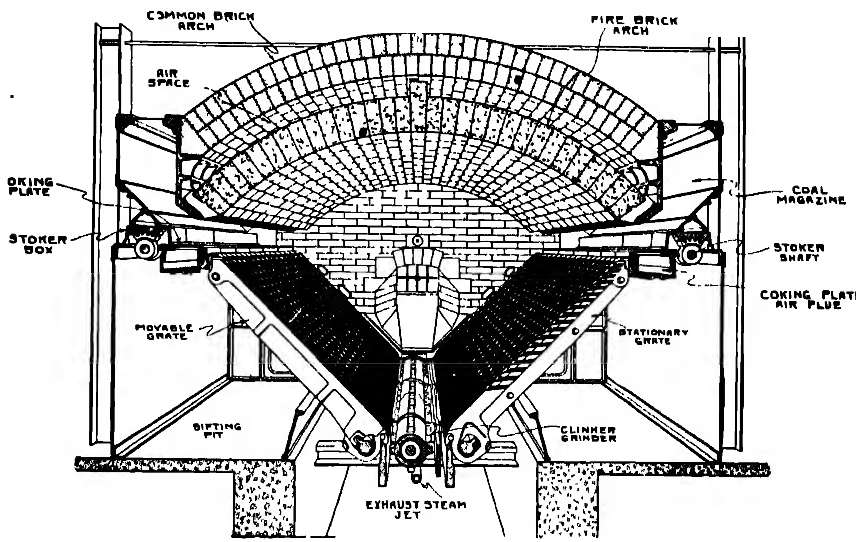


Fig. 129. Murphy Stoker and Furnace.

clinker is forced to the side or back of the fuel bed, where it is removed either by hand or by manually manipulated, or power-actuated, **dump plates.** Underfeed stokers have practically supplanted all other types in the modern large central stations burning eastern caking coals. The tendency of caking coal to swell augments the action of the stoker in producing a fuel bed of unusual thickness, and the pushing action of the feeding mechanism keeps the bed broken up and porous. The high fusing temperature of the ash and the low ash content of the eastern caking coal combine to make the cleaning periods infrequent and of short duration. With these coals, boiler ratings of 450 per cent have been realized during peak loads. All underfeeds are essentially forced-draft stokers, since they operate with restricted air openings and very deep fires. Other grades of bituminous coals have been burned successfully with underfeed stokers, but considerable difficulty is experienced with clinkers from the low-fusion ash

variety. Small sizes of anthracite, culm, and coke breeze have also been burned with some success when mixed with bituminous coal. Underfeed stokers as a rule require no ignition arches. There are two general classes of underfeeds, the **single retort** and the **multiple retort.** In practically all the former, the retorts are horizontal, while, in the latter, they are inclined.

Figure 130 shows the general principles of the **Jones "Standard"** underfeed stoker, illustrating one of the earliest and still extensively used designs of underfeed stokers of the single-retort class. It consists of a

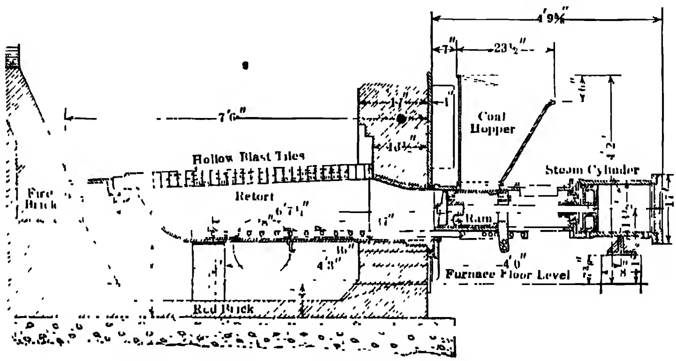


FIG. 130. Jones "Standard" Underfeed Stoker.

steam-actuated ram with a fuel hopper outside of the furnace proper and a fuel magazine and auxiliary ram within. Air for combustion is admitted through openings in the tuyere blocks on either side of the retort. Coal is fed into hoppers and forced *under* the bed of fuel in the stoker retort, where it is subjected to a coking action. After liberation of the volatile gases, the coke is pushed toward the top of the fire. The top of the fire, nearest the boiler, is always incandescent. Each charge of coal is given an upward and backward movement forcing the ash to the "dead" plates on either side of the retort from which it is removed by hand. There are no live grate bars and hence no need of an ashpit. Air is admitted through the tuyere blocks at the point of distillation of the gases. The standard size of the retort is about 6 ft. in length, 28 in. in width, and 18 in. in depth, and experience has shown that other sizes are not necessary since the spaces between retort and side wall of the various furnaces may be provided for by extending the width of the dead plates. One or more stokers are installed in each furnace, depending upon the capacity of the boiler and the width of the furnace. The steam pressure automatically controls air and fuel supply, proportioning them to each other and to varying loads in the correct degree. The result is that the stoker, if correctly installed and operated, effects complete and smokeless com-

bustion. The only variable element in the operation of this stoker, once it is correctly installed, is cleaning of fires, but if the fireman is careful to burn down the coals before breaking them up, the production of smoke may be avoided. When the fires are being cleaned, cold air rushes into the furnace and cools the setting.

Other and newer types of Jones underfeed stokers are the "**Side Dump,**" "**A-C**" and "**Lateral Retort.**" The side dump differs from the "**Standard**" only by the substitution of sloping grate bars and hand-operated dump plates for the hand-cleaned dead plates. This arrangement greatly reduces the labor of cleaning the fires. The "A-C" stoker is of the multiple-retort class and comprises a number of horizontal rams, inclined retorts, stationary overfeed sections and single dumping plates. The "Lateral Retort" consists essentially of two "A-C" stokers placed back to back in such a way that there is one main retort extending from the front wall of the furnace to the bridgewall, with the lateral retorts branching off this central retort at right angles. The "Lateral Retort" is particularly adapted to boiler units of from 100 to 500 hp.

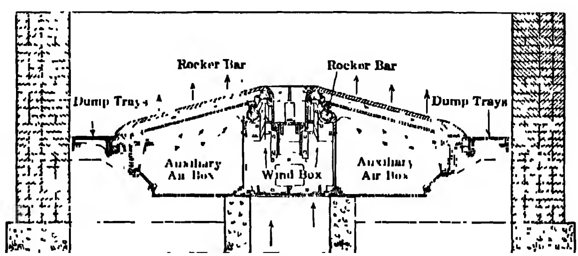


FIG. 131. "Type E" Stoker — Front Sectional Elevation.

The medium duty "**Type E**" stoker, Fig. 131, is another well-known example of the single-retort class. In this stoker the coal is fed by coal-conveying machinery or hand labor into the stoker hopper and carried under the fire by means of the reciprocating sliding bottom of the retort which runs the full length of the retort. The coal is delivered uniformly from front to rear by auxiliary pushers, and, as it rises in the retort, it is distributed to the arches of the furnace by means of moving fire bars. The fire bars move the burning fuel to the dumping tray along each side wall, where the resulting ash is deposited. The trays are dumped by a ratchet and lever on the outside of the furnace front. The coal-feeding capacity per retort varies from 200 to 9000 lb. per hr. Single retorts are used for boilers ranging from 100 to 600 hp., double retorts from 500 to 1300 hp., and triple retorts from 1000 to 2400 hp. Single-retort underfeed stokers do not require large ashpits and ash tunnels below the boiler-room floor. They are particularly adapted to installations in which more than two boilers are placed in a battery, since side doors are not necessary to their operation.

Figure 132 shows a general assembly and Fig. 133 a sectional side elevation of a **Taylor " Type H "** underfeed stoker illustrating the modern multiple-retort type. The stoker consists of a series of alternate retorts and tuyere boxes inclined as indicated. Each retort is fitted with a reciprocating piston or ram for feeding and a number of auxiliary pusher plates for distributing the fuel; it also has a movable extension grate for completing the combustion, and a dumping plate for ash disposal. The

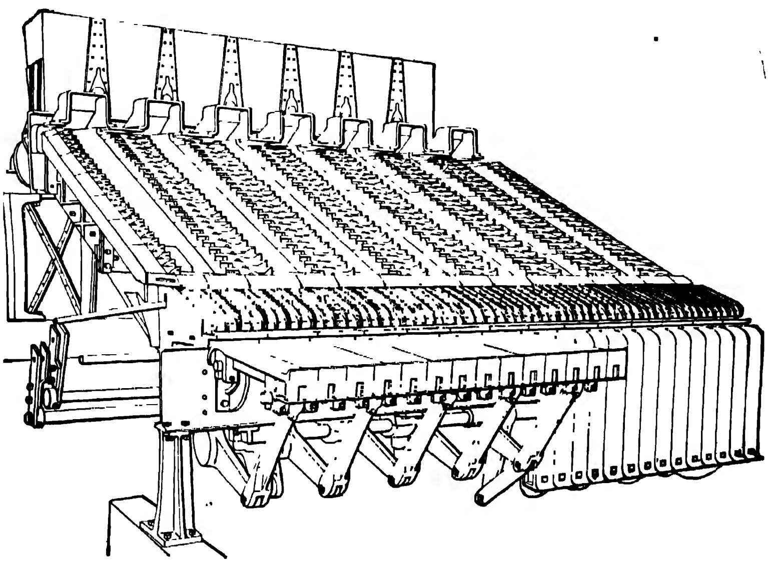


FIG. 132. General Assembly of Taylor " Type H " Underfeed Stoker.

extension grates are slowly reciprocated by the same mechanism that moves the auxiliary pushers, and the dump plates are dropped and raised by a steam cylinder. The rams and feeding system may be operated by any type of engine or motor through the conventional crank-shaft drive and gear reduction boxes or by means of hydraulic cylinders. The hydraulic drive has the merits of extreme flexibility of control with complete elimination of breakage due to foreign matter in the coal. A variable-delivery, reversible-discharge type of pump, driven by a special motor, is used to actuate the stoker. The " Type H " stoker is also equipped with rotary ash discharge or **clinker grinders** when desired. The operation of the stoker is as follows: Coal is fed into the hopper and drops behind the **feeding rams.** These rams push the coal into the top of the **retorts,** crowding upward the fuel previously introduced. Part of the green coal moves down the retorts and is pushed into the fire by the adjustable-stroke **distributing pushers.** The fuel bed is from 2 to 4 ft.

deep above the tuyeres, and, as the green coal works upward and back, it is slowly coked by the heat of the fire above. The air and gases arising through the bed of incandescent coke are thoroughly mingled and burn with an intense, relatively short flame. As the coke is consumed, it shrinks and works slowly downward, aided by the movement of the pushers underneath. Combustion of the coke is completed on the overfeed section or extension grate, from which it is forced to the dump plates. This stoker

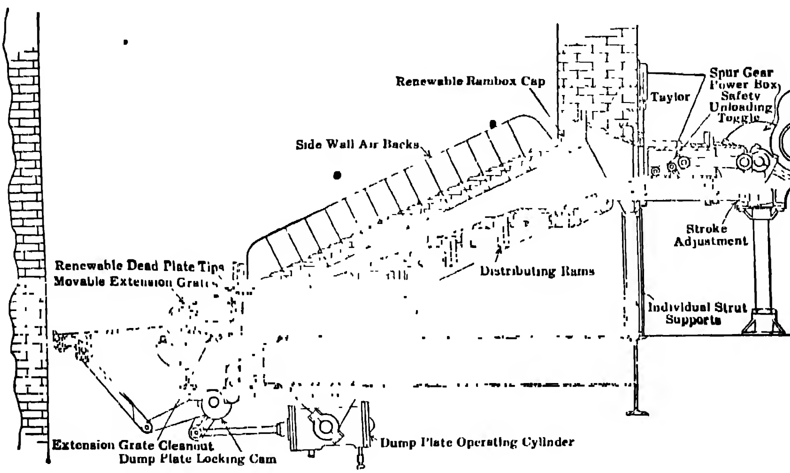


FIG. 133. Taylor "Type H" Stoker—Side Sectional Elevation.

can be built in any furnace depth from 7 ft. 8 in. to 19 ft. In the latter case, the maximum fuel-burning rate would be approximately 2500 lb. per retort.

Figure 134 gives a sectional side elevation of the new model **Westinghouse** underfeed stoker. The device consists essentially of downward inclined rams, stationary underfeed section, downward inclined adjustable secondary rams, reciprocating overfeed section and side-controlled double dumping grates. This stoker uses forced draft for its operation, and the air supply is controlled from the front. Air is admitted through the casting supporting the front wall, to the underfeed section through the tuyeres, to the overfeed section, and to the front and rear dumping grate. The rear dumping grate is replaced by a clinker grinder, where the character of the fuel and the load conditions warrant this procedure. For "base-load stations" where loads are uniform, clinker grinders are especially applicable, but where extreme flexibility is desired the dumping grates are preferable.

The **Riley "Standard"** underfeed stoker, Fig. 135, is of the multiple-retort type with an incline of about 20 deg. Instead of stationary tuyeres,

it has moving, air-supplying grate blocks, carried by the reciprocating sides of the retorts. These retort sides also move the overfeed grates, which extend across the entire width of the stoker below the retorts. Be-

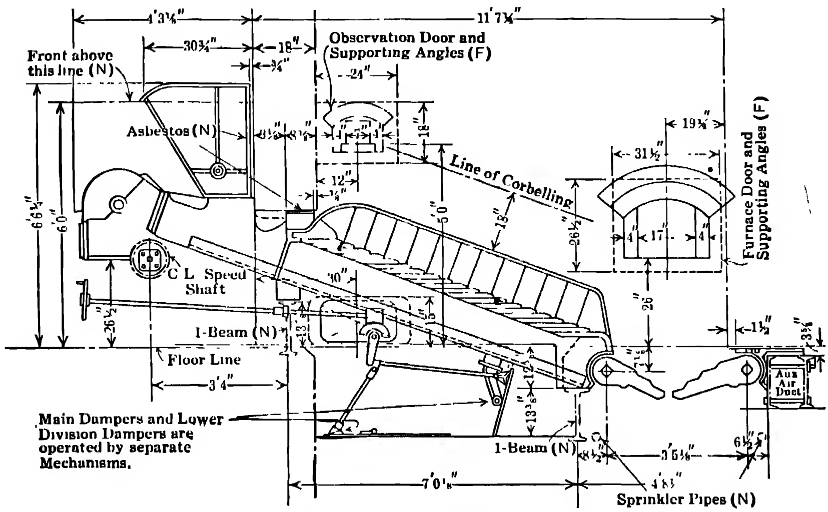


FIG. 134. Westinghouse "New Model" Underfeed Stoker.

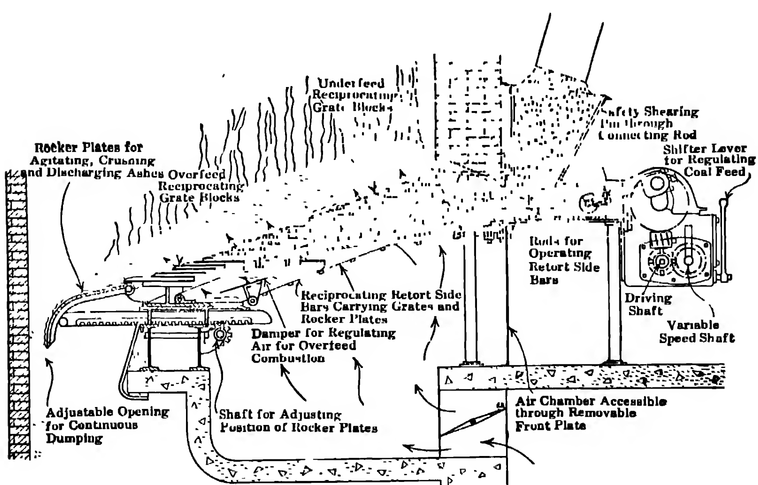


FIG. 135. Riley "Standard" Underfeed Stoker.

yond these are the rocker dump plates which continuously agitate, crush, and discharge the ash. The travel of the reciprocating parts is adjustable, so as to control completely the movement of the fuel bed and dumping

of refuse. No special shape of wind box is necessary, since the air chamber is formed by the boiler side walls and any convenient floor. Air and fuel supply may be controlled either by hand or automatically. In the older central stations with large boiler units, it was general practice to use two "standard" stokers placed opposite each other, so as to permit operation at high capacity, but the modern tendency is to install but one stoker of

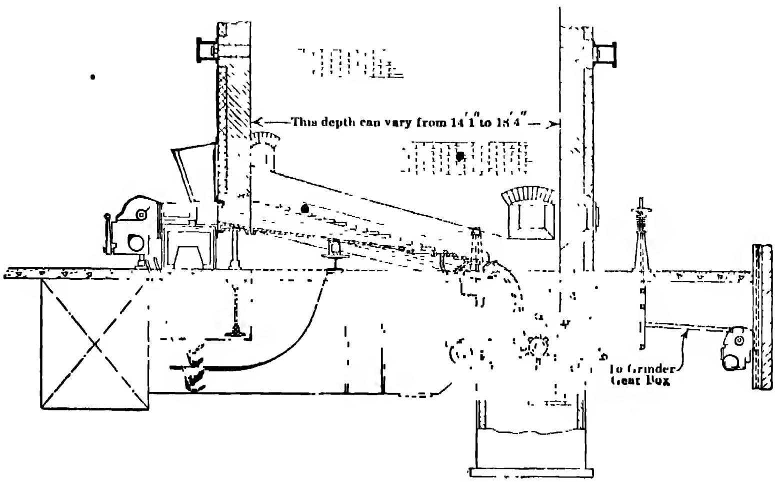


FIG. 136. Typical Installation of Riley "Super"-Stoker.

sufficient capacity to carry the load. These large stokers are frequently designated as **super-stokers.** An application of a Riley super-stoker to a large B. & W. boiler is shown in Fig. 136.

The Design and Operation of Underfeed Stokers: by H. F Lawrence, Trans. A.S.M.E., Vol. 44, 1922, p. 797; Power, Sep. 9, 1924, p. 401.

The Practical Operation of an Underfeed Stoker: Power, Jan. 31, 1922, p. 179.

Underfeed Stokers and Midwest Coal: Power Plant Engrg., Feb 15, 1924, p. 226.

112. Stoker Drives. — In order to meet the variation in steam demands, and hence the changes in rates of combustion, all stoker drives must be capable of speed variation. The variable-speed mechanism may be incorporated in the stoker itself; it may be independent of the stoker but forming the connecting link between the stoker mechanism and a constant-speed motor or engine; or it may be a variable-speed motor or engine directly connected to the stoker shaft. Because of the low speed of the feeding and stoking mechanism, there is usually a fixed speed reduction between driver and stoker shaft. The power requirements are very small and vary from 1 hp. or less in the smaller sizes of chain-grate stokers to about 25 hp. in the largest designs of underfeed stokers equipped with clinker grinders.

Because of its ease of operation and installation and wide range of speed variation, the single-cylinder piston engine, direct connected, geared or belted to the stoker shaft, is the simplest and cheapest stoker-drive obtainable provided the station heat balance permits of its use. Geared steam turbines have also been used in this connection but only to a very limited extent.

In the modern central station the stokers are usually driven by electric motors. Direct-current motors lend themselves to efficient speed regulation within rather wide limits and are to be found in many of our latest plants. The principal and only real objection to the use of direct-current motors is the matter of direct-current generation. In alternating-current stations, direct current is obtained by means of direct-current geared turbo-generators, synchronous converters, motor-generator sets, and in a few cases, from a small direct-current generator mounted on the end of the main turbine shaft. In most of the new installations, direct current is used at 230 volts, since it is easier to build adjustable-speed motors for 230 volts than for higher voltages. The usual equipment for direct-current motors for stoker drive consists of a protective panel to give overload and low-voltage protection and a drum controller and resister.

With alternating-current motors, speed variation is effected as follows:

(1) A **constant-speed** motor with a mechanical speed-changing device, such as a gear box or a variable-speed transmission, such as the **Reeves.**

(2) **A wound motor** with 2 to 1 speed control by resistance in the secondary and a 2 to 1 gear box, giving a total speed range of 4 to 1.

(3) A 4-speed **squirrel-cage** motor giving speeds corresponding to 6, 8, 12 and 16 poles, with a 2 to 1 ratio gear box making a total range of 8 fixed speeds.

(4) A 2-speed wound rotor motor giving speed of 1200 to 600 r.p.m. by pole changing with speed control by secondary resistance, thus obtaining a total range of 300 to 1200 r.p.m.

Alternating current for stoker drives is usually supplied at 440 volts.

Driving Power-house Auxiliaries: Power, Jan. 31, 1922, p. 166; May 20, 1924, p. 817.
Relation of Auxiliary Drives to Heat Balance: Power, Dec. 6, 1921, p. 888.
Control for Power Station Auxiliary Motors: Power Plant Engrg., June 1, 1923, p. 581; Power, May 13, 1924, p. 761.
A. C. vs. D. C. Motors for Stoker Drives: Power, July 3. 1923, p. 8.

113. Powdered-fuel Preparation. — Although coal may be purchased in the open market in powdered form, and custom pulverizing plants are equipped to grind lignite, peat, and other fuels upon special order, it is usually more economical to prepare the powdered product in a special plant at the point of consumption. The portion of the preparation plant

that is required for the unloading of the bulk fuel from **railroad cars**, barges, or truck to storage, and transportation from storage pile to boiler-room bunkers, differs in no way from the corresponding portion of a similar stoker-fired plant. This is also true of the removal of tramp iron, such as bolts, nuts, and pick points, by **magnetic separators**, and the crushing or granulating of the lump fuel. In either case, no preliminary crushing is necessary if the green fuel is furnished in sizes less than 1.25-in. to 0.5-in. screenings, the exact size depending upon the type and size of mill. If the crushed green fuel at ordinary room temperature contains less than 1 or 2 per cent of extraneous moisture (that which is driven off when the fuel is exposed to dry air at temperatures ranging from 86 to 95 deg. fahr. and designated by the U. S. Bureau of Mines as "air-drying loss") in addition to the so-called inherent free moisture,[1] no artificial drying is necessary, and the granulated material may be fed to the grinders directly from the green-fuel bins or storage. The inherent free moisture does not interfere with the operation of grinding, conveying, and feeding, unless the fuel has been heated to such a temperature that this moisture is vaporized and subsequently condensed upon cooling. More than 2 per cent of extraneous moisture will reduce the capacity of any pulverizer, and may seriously interfere with the conveying and feeding of the powdered product; it is therefore customary to dry all fuels in which the extraneous moisture exceeds this amount. The maximum permissible "moisture" (as ordinarily determined from the proximate analysis) for economical grinding of various fuels is substantially as follows:

	Per Cent Range	Average		Per Cent Range	Average
Western Bituminous...	3–10	6			
Eastern Bituminous....	2–8	4	Lignite	5–15	12
Anthracite.....	1–3	2	Peat	5–15	12

This is strictly applicable only to the **central or storage system**, in which the preparation of the fuel is centralized and the powdered product is stored. In the **unit system**, where the fuel is prepared as needed, and no provision is made for storing the dust, preliminary drying is ordinarily dispensed with.

Coal dryers for steam power purposes are usually of the **rotary-kiln type**, consisting of either a single or a double shell fitted with suitable rollers and gearing to permit of rotation about the longitudinal axis.[2] The shell is set at a slope of from 1/2 to 3/4 in. to the foot and so arranged

[1] "Moisture" as determined from the proximate analysis less "air-drying loss."

[2] The rotary drier is being supplanted by the waste-heat or flue gas drier and the steam drier using steam bled from the main generating unit. See Report of Prime Movers Committee, N.E.L.A., Sept., 1925, p. 43.

that the fuel being dried is subjected to the temperature of the products of combustion from an independently fired furnace. The products of combustion pass around the outside of the shell (indirect heating), through the shell and fuel (direct heating), or both around and through the shell, depending upon the type of dryer. In order to prevent overheating of the fuel in the directly-fired type, the products of combustion from the small furnace are heavily diluted with air so as to lower their temperature. The shell rotates at 1 to 3 r.p.m., and, owing to its slope, forces the fuel to gravitate from one end to the other. It requires from 30 to 50 minutes for the fuel to pass through the shell.

Figure 137 shows a general assembly of a Fuller-Lehigh dryer illustrating the **single-shell, indirectly-fired** type. The cycle of operation when burning Illinois screenings is also shown. The Bonnot dryer is a well-known example of the **single-shell, directly-fired** type and the Ruggles-Coles " Class A " of the **double-shell, directly-fired type.**

The total heat required to dry the fuel depends upon the amount of moisture to be removed, the heat absorbed by the fuel itself in passing through the dryer, the temperature difference between the air entering

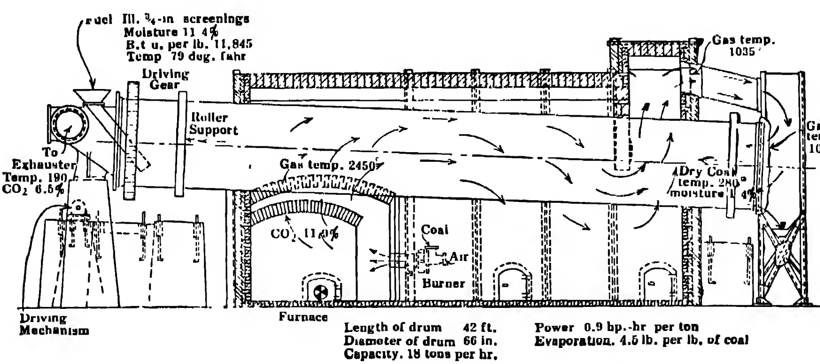


FIG. 137. Fuller-Lehigh " Indirect " Coal Dryer.

the furnace and the products of combustion leaving the dryer, radiation, and other minor losses. The overall efficiency (ratio of heat usefully applied to that supplied) of the modern coal dryer ranges from 70 to 85 per cent. This is on the assumption that the heat absorbed by the dry fuel itself is considered " useful." The overall efficiency (ratio of heat required to evaporate the water only, to that supplied) ranges from 50 to 70 per cent. A rough rule is to allow 6.5 to 7 lb. of moisture per lb. of coal as fired. The power required to operate the dryer ranges from 1.0–1.5 hp-hr. per ton for small machines having a capacity of 2 tons per hr., to 0.4 hp-hr. per ton for machines of 25 tons capacity per hour.

After the fuel has been crushed and dried (if necessary) it is conveyed to storage or directly to the **mills** where it is pulverized.

The finer the particles of fuel the more readily will they burn, and the shorter need be their path in the combustion chamber before oxidation is complete; but the cost of grinding increases very rapidly with the degree of fineness, and a point is soon reached where the gain is offset by the additional cost of preparation. In modern boiler practice employing the central or storage system, the following divisions of mesh appear to be productive of economical results for all fuels:

65	per cent	through	200	mesh
92	"	"	100	"
98	"	"	80	"
100	"	"	50	"

There are various types of grinders on the market, depending for their action upon shearing, attrition, crushing by pressure, crushing by impact, or combinations of the above. The fineness of the product is controlled by the rate of feed of raw material, screening, air separation, or combinations of these methods. A description of the various machines, involving the different principles of grinding and separation, is beyond the scope of this text, and only a few of the more commonly used types will be discussed.

Figure 138 shows a section through a **Fuller-Lehigh Pulverizing Mill** illustrating the **ball and race** type of grinder with combined air and screen separation. The pulverizing element consists of four unattached steel balls which roll in a stationary, horizontal, concave-shaped, grinding ring. The balls are propelled around the grinding ring by means of four pushers. The crushed material fed into the mill falls between the balls and grinding ring in a uniform and continuous stream, and is reduced to the desired fineness in one operation. Air is drawn into the mill at the top and, traveling downward through the center of the upper or separating fan, passes over the pulverized particles and lifts them into the chamber above the grinding zone. The lower fan acts as an exhauster and draws the dust through the finishing screen, which completely encircles the separating chamber. The material leaving the separating chamber is drawn into the lower fan housing, from which it is discharged through a spout by the suction of the lower fan. All the powdered product is discharged from the mill in finished condition and requires no subsequent screening, sizing, or separation. Speed of rotation 130 to 450 r.p.m.; the lower speeds for the larger machines. Other well-known makes of mills for pulverizing fuel are the **Raymond, Bonnot, Stroud, Allis-Chalmers, Kennedy-Van Saun,** and **Hardinge.** Mills of this general class require from 10 to 25 kw-hr. per ton of finished product, depending upon the physical properties of the

fuel and the degree of fineness desired. Powdered fuel mills, of whatever type, are seldom built with capacities over 20 tons of powdered product per hour.

Figure 139 shows a section through a **Seymour pulverizer,** illustrating the type of grinder commonly used in small "unit" installations. The

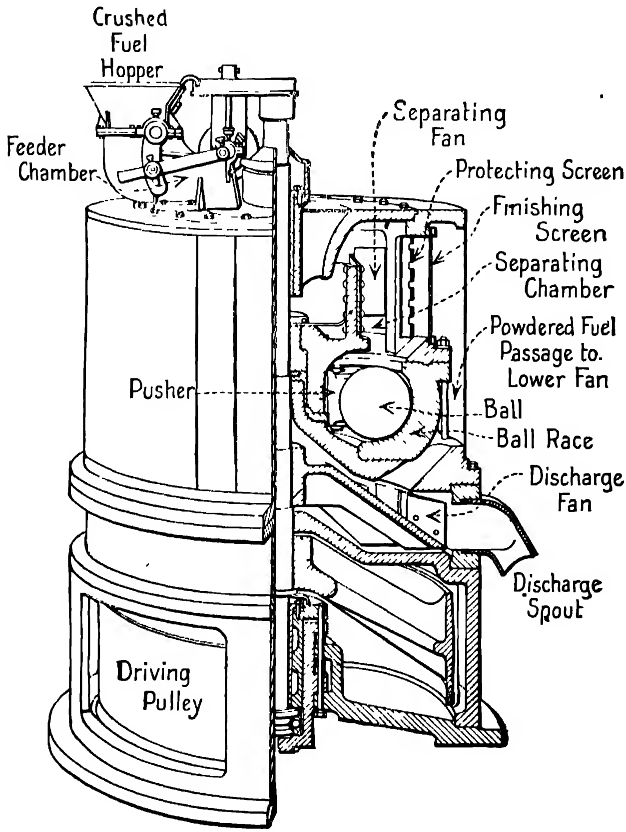


FIG. 138. Fuller-Lehigh Pulverizing Mill.

mechanism consists essentially of a cylindrical housing containing a rotating pulverizing element and a fan. The crushed fuel is reduced to powder by attrition and percussion, through the centrifugal action of the rapidly revolving vanes. The fan element draws in sufficient air to propel the fuel through the pulverizer and at the same time to support combustion in the furnace. No preliminary drying is necessary except with the very wet fuels, and screens are dispensed with entirely. This device is made in sizes ranging from 1/4 to 2 tons of powdered product per hour.

Speed of rotation for the smallest unit, 1800 r.p.m.; for the largest, 720 r.p.m.; size of driving motor, 15 hp. for the smallest unit and 60 hp. for the largest. The **Aero Pulverizer, Stroud,** and **Pulverburner** are other well-known examples of high-speed impact grinders intended primarily for small unit installations.[1]

114. Powdered-fuel Feeders, Mixers and Burners. — There are many successful systems for feeding, mixing, and burning powdered fuels, but they overlap to such an extent that a simple classification is impossible. In the small "unit" systems, grinding, feeding, and mixing are carried on simultaneously in a single housing, and it is only necessary to install the self-contained apparatus in front of the boiler setting, attach the inlet to the raw-fuel hopper and connect the discharge spout to a suitable cylindrical nozzle projecting into the furnace. Among such appliances may be mentioned the Aero-pulverizer and the Seymour Pulverizer. Preliminary drying is not necessary except with very wet fuels. Unit systems of this type require from 22 to 35 kw-hr. per ton of fuel for their

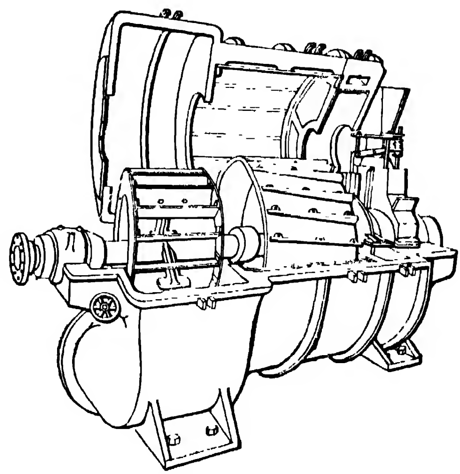


FIG. 139. Seymour Coal Pulverizer — Unit Type.

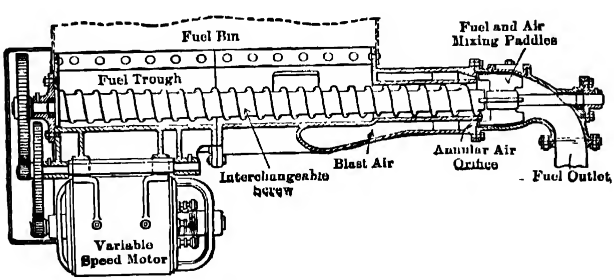


FIG. 140. "Lopulco" Powdered Coal Feeder.

operation and are made in sizes ranging from 500 to 5000 lb. of fuel per hour. A serious deficiency of this system from an operating standpoint is the lack of a reserve supply of powdered fuel, since any interruption due to breakage or the like necessitates shutting down the boiler. Peak

[1] *Development of Unit Pulverizers*· Mech. Enrg., Mid Nov., 1925, p. 1047.

demand of the pulverizer equipment is also coincident with that of the main plant.

Figure 140 shows a section through the **Lopulco feeder,** manufactured by the Combustion Engineering Corporation, and Fig. 141, a similar view of the **Lopulco induction burner.** The feeder is of the screw type, operated by a variable-speed motor. Powdered fuel is fed by the screw to a

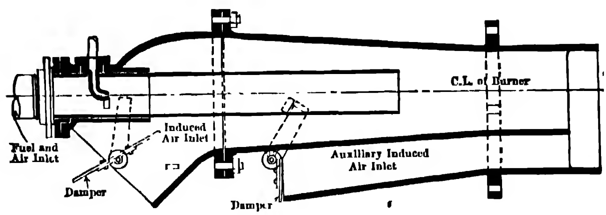


FIG. 141. "Lopulco" Powdered Coal Burner.

small mixing chamber provided with paddles, where it meets a jet of primary air supplied under a pressure of approximately 6 ounces. Through the action of the paddles and the jet, the fuel and air are thoroughly mixed before being forced into the burner. The primary jet furnishes only a small portion of the air required for combustion. The furnace vacuum sometimes extends back into the primary air pipe, while part of the sec-

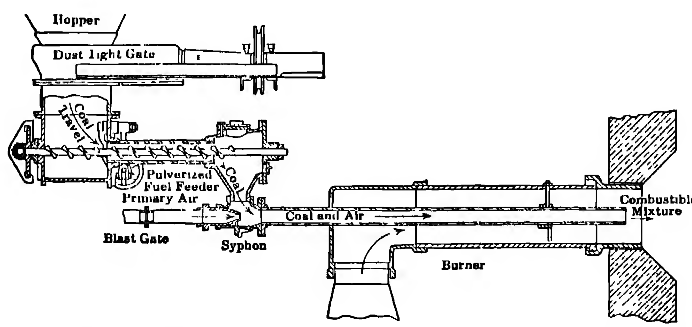


FIG. 142. Quigley Central-duct Powdered-fuel Feeder and Burner.

ondary air is admitted through a cellular casting which surrounds the fuel pipe. The remainder of the secondary air is admitted through dampers in the front wall of the furnace.

Figure 142 gives the general details of the **Quigley feeder** and burner. Powdered fuel is withdrawn from the hopper by a constant-speed screw (equipped with adjustable shutters for controlling the rate of feed) and dropped into a syphon tee, where it meets the primary air jet. The primary air is supplied by a fan or blower under a pressure of 6 to 8 ounces

and represents about 12 per cent of the air required for combustion. The jet creates a slight vacuum about the screw, preventing bridging or arching of fuel in the bin, and at the same time forces the fuel into the furnace. The secondary air is supplied by a low-pressure blower and enters the burner pipe through the annular space around the primary air line. The

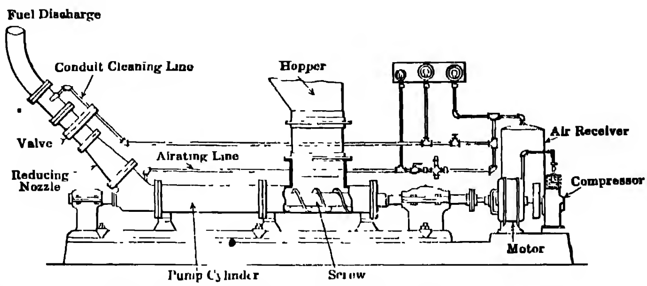


FIG. 143. Fuller-Kinyon Pulverized Material Conveying System.

expansion of the primary air by heat in the furnace causes the fuel to mix with the surrounding annulus of combustion air. The length of the flame is controlled by changing the position of the tip of the primary air pipe with respect to the nozzle of the burner. Advancing the tip delays the mixing and gives a larger flame. Rated capacities of the Quigley burners range from 200 lb. to 1800 lb. per hr.

Figure 143 illustrates the feeder, and Fig. 144, the induction-type burner of the **Fuller-Kinyon system.** The capacity of the feeder is controlled by the speed of the screw. Powdered fuel is delivered by the screw to the inner tube of the burner, where it is picked up by a jet of air supplied by a blower under a pressure of 2 1/2 ounces and projected into the furnace. This primary air supply represents approximately 50 per cent of the total required for combustion. Secondary air is induced from the atmosphere, by the action of the primary jet, into the annular space surrounding the inner tube of the burner. The tip of the inner tube does not enter the furnace but is set back from the burner nozzle as indicated. The burner is designed so that the fuel-air mixture enters the furnace at low velocity.

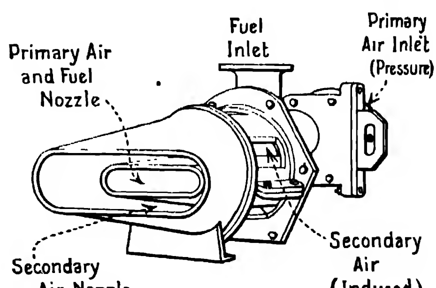


FIG. 144. Fuller Induction Burner. Vertical Type for Low-Volatile Coals.

Figure 145 gives a sectional view of the **"Multi-mix"** powdered fuel burner and feeder, which differs from other blast-type devices in the manner of

mixing the fuel and air and of introducing the mixture into the furnace. All of the combustion air is supplied through the burner, and no auxiliary air admission is required in the furnace. Because of the comparatively low feed velocity and the absence of any "scrubbing" action of the jet, smaller furnace volumes may be employed than with the high-velocity types. The drawing is self-explanatory and requires no detailed discussion.

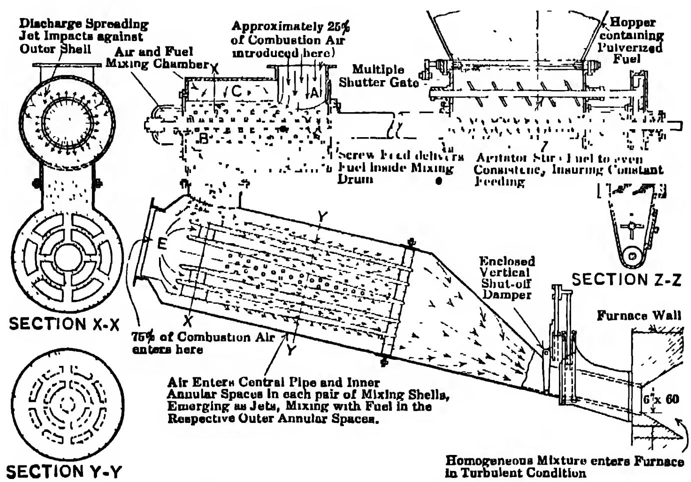


FIG. 145. "Multi-mix" Powdered-fuel Feeder and Burner.

Other popular makes of powdered-coal burners and feeding devices are the **Grindle,** manufactured by the Whiting Corporation, Harvey, Ind., the **Rayco,** Raymond Bros. Engrg. Co., Chicago, and the **Stroud,** E. H. Stroud, Chicago.

115. Powdered-fuel Furnaces. — The modern furnace for the efficient combustion of powdered fuel differs but slightly from that of an oil-fired structure, since powdered fuel behaves more nearly like a fluid or gas than it does like lump or bulk fuel. A plain chamber, without ignition arches, target walls, or deflecting arches, appears to give the best results, provided the volume is large enough to burn the fuel in suspension. The volume depends primarily upon the fineness of the fuel particles and the maximum weight of fuel to be burned per unit of time. The shape, as regards length, width, and height, is a function of the maximum permissible flame length (40 to 60 ft.); number, size, and position of the burners; provisions for cooling the furnace lining, and the type of boiler. In the latest power houses, the ratio of furnace volume to water-heating surface ranges from 0.3 to 0.85, corresponding to maximum boiler ratings of from 125 to 380 per cent. On account of the high temperatures involved, and

the slag produced from the ash, the destruction of the furnace lining is very rapid if the flame impinges directly upon the refractories. When the temperature of the furnace lining exceeds that of the molten ash, the slag projected against the lining penetrates into the brick and washes it away. If the temperature of the lining is lower than that of the slag, no erosion takes place and the slag forms a soft protective coating. The temperature of the lining is maintained below that of the slag by exposing a large portion of the boiler-heating surface to direct radiation from the furnace and by cooling the walls either with air or with water jackets. At ratings up to 200 per cent, with fuels having high-fusion ash, no slagging occurs and the refuse may be raked out in the usual way; at higher ratings or with low air excess and low-fusion ash, considerable objectionable ash is formed, but the ill effects may be largely eliminated by cooling the bottom of the furnace either through air cooling or by means of water screens. In the very latest designs, the furnace side walls are either of solid brick protected with a water-cooled surface, or of composite construction protected with a combination air and water-cooled surface. The water cooled construction is practically imperative

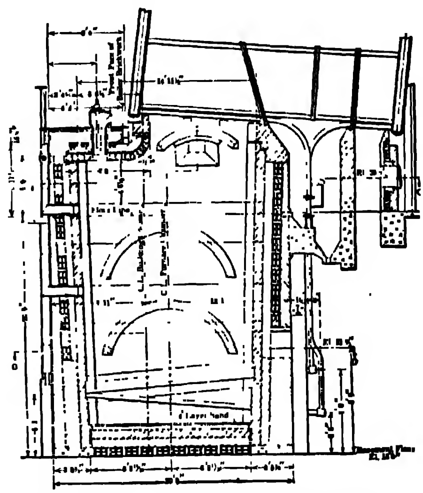


Fig. 146. Powdered-coal Furnace. Lakeside Station.

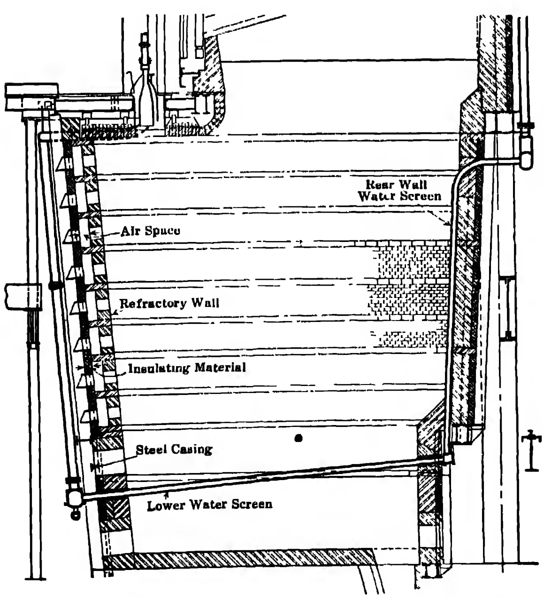


Fig. 147. Powdered-coal Furnace, Combustion Engineering Corporation.

with pulverized-fuel firing in combination with air heaters because of the intense heat in the furnace at very high ratings. The water coolers consist usually of 4-in. steel tubes arranged on each side of the furnace and spaced on 6 to 8-in. centers. The water in the coolers forms part of the regular boiler circulation. In the "fin" type of construction the spaces between adjacent tubes are covered by two steel strips or fins welded longitudinally to the tube and diametrically opposite each other,

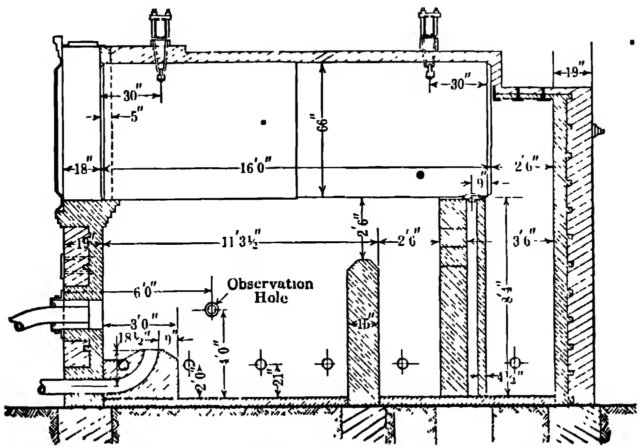


Fig. 148. Furnace for Seymour Pulverizer. Unit System.

thus presenting a continuous metal surface to the fire. For a detailed description of a number of water-cooled furnaces, consult "Water Cooled Furnaces," Mech. Engrg., Mar. 1925, p. 197.

Figures 146 to 148 give the general details of typical powdered-fuel furnaces.

Susquehanna Station of the Metropolitan Power Co.: Power, Dec. 29, 1925, p. 1000.

Cost of Preparing and Delivering Powdered Coal to the Furnace: Bureau of Mines, Bul. 217, 1923, p. 100.

Pulverized Coal: Serial Report of Prime Movers Committee, N.E.L.A., Sept., 1925. Power, June 3, 1924, p. 900. (Serial.)

Combustion Steam Generator: Power, Feb. 2, 1926, p. 232.

Turbulent Flow or Well Type Furnace: Report of Prime Movers Committee, N.E.L.A., Sept., 1925, p. 40.

116. Fuel-oil Burners. — The name "oil burner" is a misnomer, because the so-called burner does not burn the oil but merely atomizes it. The atomization may be effected by high- or low-pressure compressed air or steam, a combination of air and steam, or by merely mechanical means. For high-pressure steam generation in stationary plants, only two basic

types need be considered: viz., **steam burners,** and **mechanical burners.** The essentials of any type are (1) the complete atomization of the oil without clogging, fouling, or "drooling"; (2) jet of such shape as to insure intimate mixture at all points with the incoming air; (3) capacity for effecting complete combustion with minimum excess air at the various steaming rates contemplated; and (4) accessibility, minimum attention, and low maintenance. Neither type is of universal application; in some cases the steam atomizer is the better investment, and in others the mechanical atomizer offers more advantages. In the steam burners, the oil is atomized and forced into the furnace by a steam jet; in the mechanical burners, the oil under pressure breaks into a fine spray on passing through specially designed orifices.

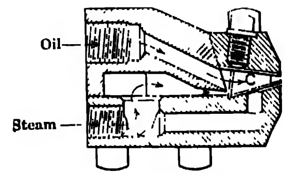


FIG. 149. Hammel Oil Burner Head (Steam Atomizer).

Steam burners are designated either as **outside mixers,** in which the oil and steam meet outside the burner nozzle; or **inside mixers,** in which the oil and steam mingle inside the nozzle. The **Hammel, Enco, Airoil, Leahy, Rogers-Higgins, Peabody, Kirkwood,** and **Tate** are representative of the inside mixers; and the **Best, Gilbert** and **Parker, Rockwell,** and **Foerst** of the outside mixers. Some of the well-known steam burners are illustrated in Figs. 149 to 157.

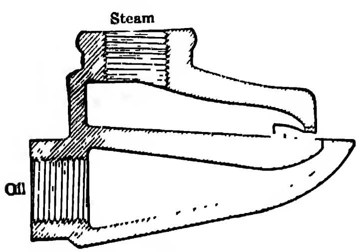


FIG. 150. Foerst Oil-burner Tip (Steam Atomizer).

Figure 149 shows a section through the atomizing tip of a Hammel burner, illustrating a well-known design of the inside-mixing type. The

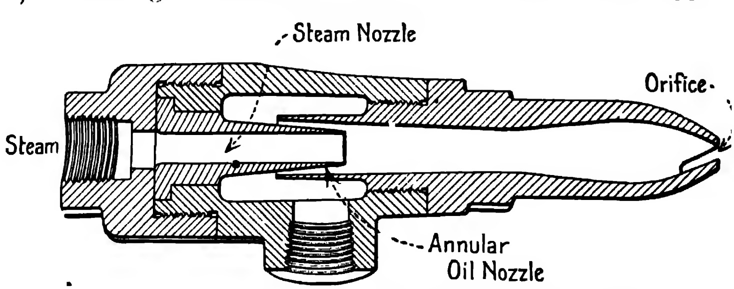


FIG. 151. National "Airoil" Burner (Steam Atomizer).

oil, under pressure and either heated or cold, is fed through the upper pipe into the mixing chamber C, where it encounters the steam jet issuing from

the lower pipe, and the mixture is forced through the rectangular orifice in the shape of a long, flat spray.

Figure 150 shows a section through the atomizing tip of a **Foerst** burner, illustrating the outside-mixing type. The oil is fed through the lower pipe at right angles to the steam jet and is discharged in the shape of flat or **fan-tail** spray.

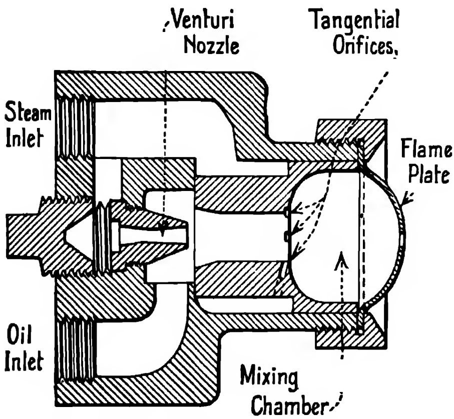


FIG. 152. "Enco" Oil Burner (Steam Atomizer).

Figure 152 shows a section through the **Enco** steam atomizer, which differs considerably from the usual type of steam atomizers in that it is suitable for either natural or forced draft. Referring to the illustration, steam and oil enter the device as indicated. Part of the steam passes through a Venturi nozzle on the center line of the burner, and the rest enters the mixing chamber through tangential openings. Oil enters around the mouth of the Venturi, is caught up and partly atomized by the center steam jet and is carried forward through the center passage to the opening of the mixing chamber. Here it is "cross cut" by the tangential jets, whirled around at high velocity in the body of the mixing chamber, and finally discharged through the orifice plate. As the oil is completely atomized inside the burner, it may be discharged through any number of openings, of any shape, at low velocity. The atomizer, complete with forced-draft air registers, is shown in Fig. 153.

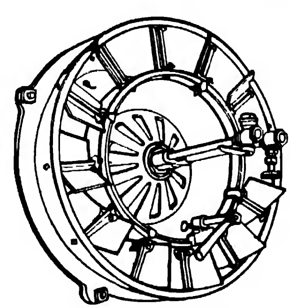


FIG. 153. Air Register for "Enco" Oil Burner.

Mechanical atomizers for high-pressure boilers are practically all of the oil-pressure type; that is, the oil is forced under pressure through suitable orifices or tips which break it up into a very fine spray, or "fog." In low-pressure installations, the centrifugal atomizer is commonly used. In this type the oil is broken up into a spray by the centrifugal action of a motor-driven rotating tip.

Figure 154 gives the general details of the **Peabody-Fisher** burner. All of the oil enters the atomizing chamber through heavy burner-tube A, and enters the atomizing chamber in tip E through several small passages

C, which give the oil the necessary whirling motion. By opening a valve at the end of inner tube *B*, which connects with a series of holes in the burner tip, part of the oil in the tip is by-passed. This design enables the same amount of oil to reach the tip regardless of the load, so that there is no reduction in the oil pressure and the spray effect is practically constant for a large flow of oil. The by-passed oil is returned to pump suction or storage.

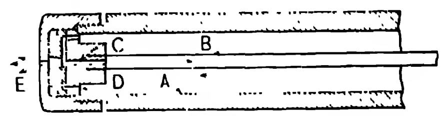


FIG. 154. Peabody-Fisher "Wide Range" Atomizer (Mechanical Type)

Figure 155 shows a section through the atomizing tip of the **Koerting** mechanical atomizer. In this device, the oil is forced under pressure tangentially through an annular chamber in the burner tip, the chamber being so arranged as to give the oil a high velocity of rotation, and thus, under the action of centrifugal force, to break it up into a fine spray. The annular chamber or tangential groove discharges into a small cylindrical chamber, the top end of which is conical in shape. The discharge orifice is at the apex of the cone.

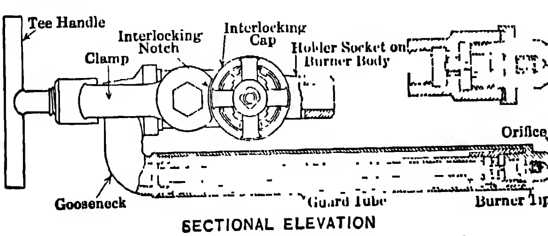


FIG. 155. Koerting Mechanical Oil Atomizer.

Figure 156 shows a section through a **Coen** mechanical atomizer, illustrating the single-orifice type in which no rotational motion is imparted to the oil before it leaves the tip. Other well-known makes of mechanical atomizers are the **B. & W. " Lodi "** and **" San Diego," White, Dahl, Fess,** and **Witt.**

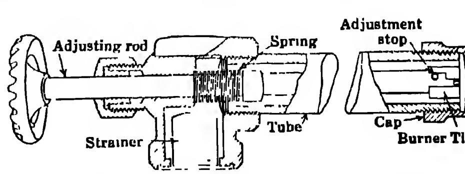


FIG. 156. Coen Mechanical Oil Atomizer.

Steam atomizers for boiler furnaces operating under natural draft are usually designed to give a flat flame and are installed in the furnace so that the flame is spread out over a floor of refractory material, as in Fig. 159. When they are provided with orifices which produce a hollow conical jet, air registers, as illustrated in Fig. 160, are necessary for efficient mixing of the air and atomized fuel. Steam atomizers, in connection with air registers, are also suitable for forced draft. Mechanical atomizers are almost always provided with air registers. Air registers are fully as important from the standpoint of design and construction, and have quite as great an

influence upon the securing of perfect combustion, as the atomizer itself.

The amount of steam required to atomize oil varies with design of burner, and the method of control. In large central stations where the burners are automatically controlled, or where they receive consistently good attention and care, the average steam consumption ranges from 0.12 to 0.25 lb. per lb. of oil fired. This corresponds to approximately 0.8 to 1.8 per cent of the total steam generated. But for regular, everyday operation in the average plant, the amount ranges from 0.3 to 0.8 lb. per lb. of oil fired, corresponding to 2 to per cent of the total steam generated. See Fig. 158.

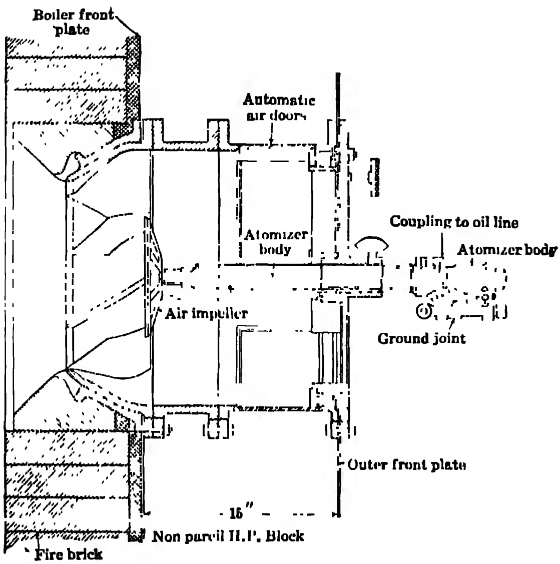


Fig. 157. B. & W. "San Diego" Mechanical Atomizer and Air Register.

The equivalent steam consumption of the mechanical burner depends

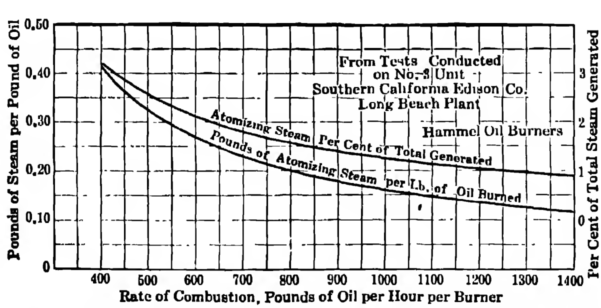


Fig. 158. Typical Performance Curves — Steam Atomizers.

largely upon the oil pressure maintained, the efficiency of the oil-pumping and heating apparatus, and whether or not the exhaust steam is utilized. In some of the latest plants, the equivalent steam consump-

tion is approximately 0.5 per cent of the total generated. As a general rule, the steam consumption will not exceed 1 per cent of the total generated.

Steam atomizers owe their popularity to the relative ease of manufacture, simplicity of installation, and the very high overall boiler and furnace efficiency realized at normal boiler rating. The burners, when properly installed, require little attention, and one man can readily control a large number of burners. For relatively large plants, an automatic control system may be installed, which regulates the burners and dampers according to the load demands so that the labor item is practically *nil* except as required for watching and cleaning the burners.

In spite of the seeming advantages of steam atomization, there are a number of factors which may prove objectionable: viz., (1) the noise made by the steam issuing from the burner, (2) difficulty and loss of time in cleaning burners, particularly in "back shot" installation, (3) blowpipe action of the flame in combustion chambers of limited capacity, (4) extra amount of moisture in the flue gas, which may prove troublesome in connection with economizers, (5) cost of steam used for atomizing, and (6) limited range in overload capacity of the boiler. The majority of steam-burner installations reach their maximum commercial capacity at boiler ratings of 175–200 per cent, although some of the latest designs in connection with forced draft and air registers have been operating satisfactorily at 300 per cent rating. Boiler ratings of 300 per cent have been maintained in modern mechanical burner plants[1] with overall efficiencies of 80 per cent (without economizers); and in special tests,[2] boiler ratings of 630 per cent have been reached with overall efficiencies of 76.6 per cent.

Steam atomizers usually operate with natural or indirect draft, while the mechanical atomizers generally necessitate the use of forced draft, although from 100 to 175 per cent rating may be secured with natural draft.

Steam atomizers for boiler service are seldom designed to pass more than 800 lb. of oil per hr. per burner, whereas single mechanical burners have satisfactorily handled as much as 1500 lb. per hr.

See also paragraph 126.

Burning Boiler Oil: Power, Aug. 7, 1923, p. 209 (Serial).
Mechanical Atomization of Fuel Oil: Power Plant Engrg., Nov. 1, 1923, p. 1130.

117. Fuel-oil Furnaces. — Because of the ease with which oil fuels can be atomized and brought into intimate contact with the air for combustion, the furnace may be of the simplest construction. No grates, ashpits,

[1] Savannah Electric Co. Test of May 16, 1920.
[2] League Island Navy Yard. Competitive burner test, 1918.

ignition arches, or target walls are necessary, and a properly proportioned plain chamber fulfills all requirements. The correct proportion of this simple furnace, however, is dependent upon a large number of factors, such as type of boiler; arrangement of tubes and baffles; elevation of headers or their equivalent above the floor lines; number, type, and location of burners; length of furnace with respect to flame travel; method of admitting and controlling air; draft; character of loads; and whether or not the furnace is to be used solely for oil, or simultaneously or alternately with other fuels. The volume of the combustion chamber in the latest oil-fired plants with high settings ranges from 0.2 to 0.4 cu. ft. per sq. ft. of boiler-heating surface, and in low settings from 0.10 to 0.15

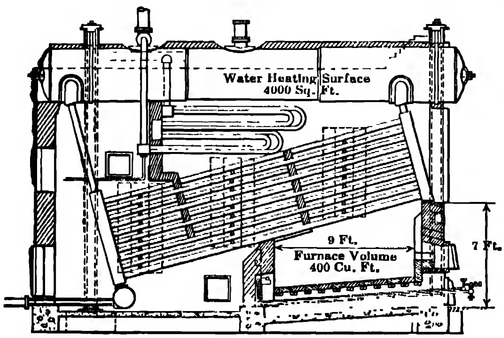


FIG. 159. Hammel Oil Burning Furnace — Low Setting, Steam Atomizer.

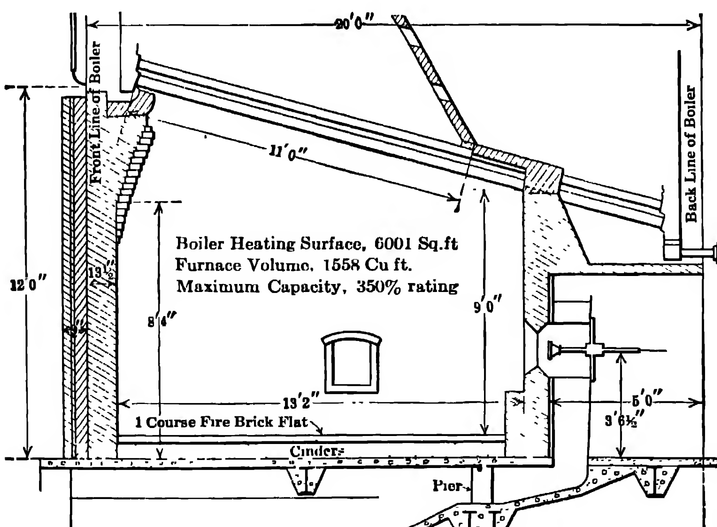


FIG. 160. Oil-fired Furnace. Narragansett Elec. Lt. Co., Mechanical Burner.

cu. ft. A rough rule is to allow 1 cu. ft. of combustion space per b.hp., to be developed. The best results have been obtained where the flame travel is complete without impingement on tubes, walls, or floors, and where as much of the boiler-heating surface as possible is exposed to

radiant energy. In general, the larger the furnace and the further the burners are from the tubes or heating surface, the better will be the results secured and the greater the overload capacity.

Steam burners of the flat-jet type almost invariably operate under natural draft, while forced draft is the more common with mechanical burners, particularly when heavy overloads are desired. The number of burners depends upon the size and type of burner and furnace, and the ratings at which it is desired to operate the boiler. In order to maintain an even distribution of the flame, a multiplicity of burners is preferred in all but the smallest boilers. It is customary, when possible, to install the burners in the front wall of the setting and project the flame toward the rear ("front shot"), but it is frequently desirable to install them in the bridgewall, and project the flame toward the front ("back shot"). With boilers of the inclined-tube type, the latter arrangement minimizes flame impingement on the tubes and affords increased furnace volume in the direction of the flames. With steam burners of the flat-jet type, it is current practice to introduce the air into the furnace partly around the burner and partly through slots in the furnace floor. With the mechanical burners, and steam burners giving a conical flame, all the air for combustion is admitted through air registers and diffusing vanes surrounding the atomizer, and at no other place.

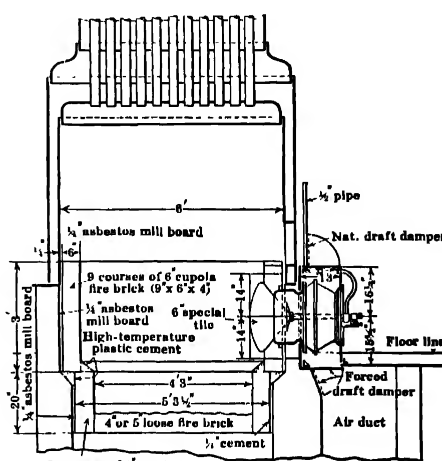


FIG. 161. Peabody-Fisher Burner as Applied to a 1500 Sq. Ft. Manning Boiler.

A modern **low-set** furnace of the **back shot** type as applied to a water-tube boiler is illustrated in Fig. 159. The burner tips are housed in slots located in the back of an arched recess in the bridgewall, and the flame is projected forward toward the front of the furnace. The furnace floor is solid except for narrow air slots through the deck and in front of each arch. Each burner with its accompanying recess has a separate air tunnel leading from the boiler front; these tunnels do not communicate with each other under the furnace floor, so that by closing the air-admission door any tunnel can be sealed up while the others are supplying air to their particular burners. The air entering the furnace is heated by contact with the incandescent floor of the furnace, the floor constituting the

roof of the air tunnels. Usually one burner and air tunnel is furnished for each 4 ft. of furnace width. This low-set type of furnace is not intended for boiler ratings over 175 per cent.

Figure 160 gives the general dimensions of the oil-fired furnace at the power house of the Narragansett Electric Lighting Co. as applied to a 600 hp. B. & W. boiler, and illustrates a modern "high-set" installation.

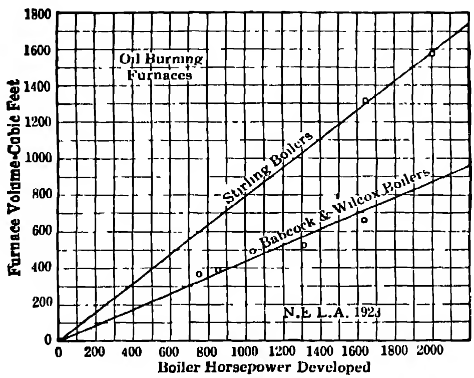


FIG. 162. Furnace Volume vs. Boiler Rating.

Figure 161 shows the application of a mechanical burner to a vertical fire-tube boiler.

Figure 162 shows the relation between furnace volume and maximum hp. developed with 14 to 15 per cent of CO_2 and no CO.

Burning of Liquid and Gaseous Fuels: Report of Prime Movers Committee, N.E.L.A. 1923, (Part B), p. 297.

Computing Guaranteed Stoker Efficiency: Power, May 20, 1924, p. 813.

The Storage and Handling of Fuel Oil in Industrial Plants: Mech. Engrg., Nov. 1924, Part 2, p. 771.

Oil Burning in Industrial-Plant and Central-Station Service: Mech. Engrg., Nov. 1924, Part 2, p. 849: Apr. 1925, p. 276.

CHAPTER VII

FUEL AND ASH CONVEYING SYSTEMS

118. Storage of Fuels. — The cost of fuel and its delivery into the furnace are usually the largest items in the operating charges; hence, large central stations are located, when practical, at or near the mine mouth and adjacent to a railway line or water front in order to insure a continuous supply of fuel and to minimize the cost of storage and handling. Isolated stations in the business districts of large cities are generally unfavorably situated, with the result that fuel storage is limited to a very small quantity and the cost of conveying is a large percentage of the total fuel cost. Wherever the plant may be located and whatever may be the system of transportation and conveyance, provision should be made, if possible, for storing a certain quantity of fuel as a precaution against interrupted delivery and possibly enforced shut-down. The amount to be stored depends upon the character of the fuel itself, method of transportation, space available, size of plant, and the cost of interrupted service. In some of our largest central and isolated stations, the fuel requirements are such as to necessitate immense piles of coal or reservoirs of oil for even a few days' operation. There are several such stations burning 3000 tons of coal or more per day. One week's supply at a rate of 3000 tons per day would occupy a space about 200 ft. square and 21 ft. high. The space occupied by fuel oil of equivalent heating value would be approximately 65 per cent of that occupied by the coal.

The weight of a cu. ft. of coal varies with the percentage of fine and coarse particles in the mass, the moisture content, and the packing effect to which it may be subjected. The variations in weight due to fineness and moisture content are less than is ordinarily supposed. A mixture of coarse and fine, such as is usually found in stoker sizes, will remain very uniform in weight per cu. ft. with reasonable changes in the percentage of moisture. The values in Table 34 are approximate only.

For data pertaining to the weight of different classes of fuel, consult "Specific Gravity Studies of Illinois Coal," Univ. of Ill., Bul. No. 89, July 3, 1916; and "Weight of Various Coals," Bureau of Mines, Tech. Paper 184, 1918.

TABLE 34

SPACE OCCUPIED BY BULK FUEL*

Bituminous	Cu. Ft. per Ton	Anthracite	Cu. Ft. per Ton
Pocahontas Lump and Egg	35.5	Chestnut	34
Pocahontas Mine-run and Nut	36	Range	35
Pocahontas Slack	35	Small Egg	35
Hockings	41	Large Egg	36
Screenings	40	Pea	33
Indiana Lump	41	Buckwheat	32
Mine-run	36	Dust	35
Smithing	43		
Quaker Egg and Nut	40	*Coke*	
Quaker Lump	38		
Acorn Lump	40		
New Era	38	Petroleum	72
No. 3 Washed Nut	42	Gas House	66
Wasco Lump	40	Solvay Nut	55

* Peabody Coal Company.

Anthracite and practically all kinds of bituminous coal have been stored without spontaneous combustion taking place, yet, under certain conditions, spontaneous combustion has occurred with every kind of coal stored. The geological age of coal is a fair guide to its liability to heat, anthracite being the safest to store and lignite the most dangerous. The spontaneous combustion of coal is largely due to oxidation of the fine sizes; consequently, the liability to such combustion in stored coal is greatly reduced, and in many cases eliminated, if dust and fine coal can be kept out of the pile. The fire hazard for piles of clean-sized coal is relatively small compared with that for piles of screenings or mine-run. Most of the fires recorded have occurred within ninety days after the coal was placed in storage. The percentage of fires in piles of mixed coal is considerably greater than in piles of the same coal unmixed. Coals that are known to be particularly liable to spontaneous combustion should not be selected for storage if it is possible to avoid doing so. Coal should be heaped for storage so that any part of the pile can be moved promptly if necessary. The arrangement should be such that air cannot enter the pile. A number of small piles is to be preferred to one large pile, but space limitations usually prevent the spreading out of the fuel. The depth of the pile depends upon the kind of coal and local conditions, but experience shows that fires are more common in deep than in shallow piles. Coal from different fields should not be stored together. Underwater storage is the only positive insurance against spontaneous com-

bustion, since not only is the fire hazard completely eliminated, but the fuel does not depreciate as when exposed to air. Space requirements, first cost of reservoirs, and special requirements for handling are factors which must be considered in this connection. For a description of several under-water storage systems, consult " The Storage of Bituminous Coal," by H. H. Stock, Univ. of Ill., Bul. No. 27, 1918, pp. 86–106.

Among the many systems of unloading coal from cars or barges, and of storing it, may be mentioned the following:

- Hand-operated storage systems
- Storage by motor truck
- Pile storage from cars without trestle
- Trestle storage
- Storage with side dump cars
- Side-hill storage
- Use of mast and gaff
- Use of cable drag scraper
- Locomotive crane
- Gasoline and steam-operated caterpillar crane
- Revolving car dumper
- Parallel track storage
- Trestle and traveling crane
- Circular storage
- Steeple towers
- Bridge storage or gantry crane
- Deep reinforced-concrete bins
- Skip hoist and monorail
- Under-water storage
- Silo-type concrete and vitrified-tile bins

A description of these various systems and the methods adopted for detecting spontaneous combustion are beyond the scope of this text, and the reader is referred to that excellent treatise " Bituminous Coal Storage Practice," by Stock, Hippard and Langtry, Univ. of Ill., Bul. No. 2, Jan. 19, 1920. See also, Report of Prime Movers Committee, N.E.L.A., March, 1925.

A certain amount of coal should be stored, if possible, within the station itself. In the smaller plants it is customary to place the coal in an open pile in front of the boilers or in a bin below, or on the same level with the boiler-room floor. In the larger plants, it is common practice to install **overhead bunkers** so that the fuel can be fed to the furnace by gravity through down spouts. In some of the latest central stations, additional storage is provided by pits underneath or alongside the car tracks and within the main building itself. See Figs. 185 and 186. Overhead bunkers are rectangular or circular in plan and are built of steel plates lined with concrete, refractory materials, or reinforced concrete. The bottom slope should not be less than 45 degrees to the horizontal. **Suspended bunkers** of reinforced concrete construction are also in evidence in modern plants. With certain grades of fuel, separate bunkers for each boiler are preferred to a large single container for the entire plant, since fire resulting from spontaneous combustion is more readily prevented

from spreading. **Silo-type bins** of concrete and vitrified tile are finding favor with many engineers where the quantity of coal to be stored is not very large.

Fuel oil is stored in covered steel tanks, weather-proofed concrete tanks, or earthen reservoirs. Steel tanks are used in the great majority of the plants, but excellent results have been reported from users of concrete tanks. An earthen reservoir of 25,000 barrels' capacity at the upper plant of the San Joaquin Light & Power Co. is said to show a loss of but 100 barrels per month through evaporation and seepage. Steel tanks may be placed entirely above the ground or they may be partly or completely buried, depending upon plant location, Underwriters' requirements, and community ordinances. Concrete tanks are generally installed below the ground. For detailed description of the various systems of storage and for a brief outline of the rules and requirements of the National Board of Fire Underwriters for storage and use of fuel oil, consult the reference at the end of this paragraph.

Factors in the Spontaneous Combustion of Coals: Mech. Engrg., Dec. 1923, p. 691.
Pipe Line Transmission of Crude Oil: Power Plant Engrg., Dec. 1, 1919, p. 1045.
Fuel Oil Containers and Tanks: N.E.L A., T3-1922, p. 254.
Fedco Protectometer Systems for Stored Coal: Power, Nov. 27, 1923, p. 850.
Fuel-oil Storage Rules, National Board of Fire Underwriters: Power, Nov. 4, 1919, p. 680.
Oil Storage and Reservoirs: C. P. Bowie, U. S. Bureau of Mines, Bul. No. 155, 1918.

119. Methods of Handling Bulk Fuel and Refuse. — The best method of delivering bulk fuels from storage to the furnace and of removing refuse from the ashpit is the one which will effect the desired result at the lowest ultimate cost. That this problem does not offer a simple solution is evidenced by the many diversified combinations found in practice for the same operating conditions. The principal factors which influence the choice of system are size and location of plant and cost of fuel and labor. In many plants, continuity of operation may be of even greater importance than reduction of cost, and extra investment may be considered advisable to offset the unreliable labor element. Of the various methods found in current practice, the following are the more common:

1. Hand shoveling.
2. Wheelbarrow or industrial car and shovel.
3. Continuous conveyors:
   - Spiral or screw
   - Scraper or flights
   - Apron and buckets
   - Overlapping pivoted buckets
   - Endless belt.
4. Pneumatic systems:
   - Pressure blowers
   - Exhausters.

5. Hoists.
   - Skip hoist
   - Grab bucket
   - Traveling crane
   - Jib and bracket crane
   - Monorail telpher
   - Platform elevator
   - Continuous bucket elevator.
6. Hydraulic systems:
   - Open trench
   - Closed conduit.

Coal Conveyors for Modern Power Plants: Power Plant Engrg., Sept. 15, 1923, p. 917.

120. Hand Shoveling. — Where possible, the coal is dumped direct from the cars or wagons into bins located in front of the boilers. In such instances one man may handle the coal and ashes and attend to the water level of 400 hp. of boilers equipped with common hand-fired furnaces. This refers, of course, to average good coal not too high in ash nor productive of much clinker. With hand-shaking and dumping grates, 500 hp. may be fired by one man, and from 1800 to 2500 hp. with automatic stokers into which the coal is fed by gravity. Sometimes the coal cannot be stored in front of the boilers, but must be hauled by wheelbarrow, cart, or industrial car. For distances over 100 ft. and quantities over 20 tons per day, the cost of handling the coal in this way may justify the installation of an automatic conveyor system. Hand-fired furnaces and manual handling of coal and ashes are usually associated with small plants of 500 hp. and under, but a number of large stations are operated in this way with apparent economy.

Large plants, however, are generally equipped with conveying machinery, not so much because of the possible reduction in cost of operation, taking into consideration all charges fixed and operating, but because of the large and often unreliable labor staff with which it dispenses. Hand shoveling is sometimes necessary even with modern unloading devices and drop-bottom cars, on account of the poor dropping mechanism and the freezing of coal in the cars. This is particularly true of washed coals, and it is not unusual to have an entire carload solidly frozen. In this case, it has to be picked, and even dynamited, and shoveled by hand, or the unloading tracks must be equipped with steam pipes and outfits for thawing purposes. A good man is capable of shoveling 40 to 50 tons of coal in eight hours when unloading a car, provided it is only necessary to shovel the coal overboard or through side openings. An average figure for handling coal by barrow and shovel is not far from 3.5 cents per ton per yard up to the distance of 5 yards, then about 0.25 cts. per ton per yard for each additional yard. The cost of handling coal and ashes in the small plant not equipped with conveyors or hoists varies within such limits that average values are without purpose. In twenty-five Chicago plants of this class, the cost in 1923 ranged from \$1.00 to \$2.50 per ton.

121. Continuous Conveyors. — Until quite recently, the most popular method of automatically handling coal and ashes was by means of continuous conveyors and elevators. While certain types of these conveyors are still used in the modern power house, the tendency is to do away as much as possible with machinery the parts of which are subject to excessive wear and high maintenance costs. For example: Horizontal conveyors depending upon links for their operation are being supplanted by cable drives, and bucket elevators are giving way to simple cable-operated hoists. Then, too, ashes and fuel are handled separately, because of the abrasive action of the ashes, instead of being transported by the same systems. Continuous conveyors may be grouped into two general classes:

1. Those which push or pull the material, but in which the weight of the load is not borne by the moving parts.
2. Those which actually carry the load.

A few of the more important types will be treated briefly.

Screw or Spiral Conveyors. These consist of a stamped or rolled sheet-steel spiral secured by lugs to a hollow shaft (usually a standard or extra-heavy pipe) revolving in a trough or enclosed conduit which it fits approximately. Standard sizes range from 3 to 18 in. in diameter and are made in sections from 8 to 12 ft. long.

TABLE 35

SPEEDS AND CAPACITIES OF SCREW CONVEYORS

(Fine Coal)

Diam. screw, in..	6	7	8	9	10	12	14	16	18
Maximum r.p.m. .	115	110	105	100	95	90	85	80	75
Capacity per hr. fine coal, tons..		7	14	16	21	36	48	80	110

On account of the torsional strain on the shaft, the maximum length seldom exceeds 100 ft. Single sections may be used as feeders on inclines up to 15 deg. Vertical screw conveyors are used for conveying certain materials, such as grain, cottonseed, fuller's earth, and the like, but not for bulk coal or ash. Low first cost, compactness and adaptability to space requirements are the advantages of this type, but these may be offset by high power consumption and excessive wear. The following equation gives a means of approximating the power requirements for horizontal runs.

$$\text{Hp.} = CWL/33{,}000 \tag{61}$$

in which

C = 2.0 to 3.0 for coal and 2.5 to 4.0 for ashes,
W = capacity in lb. per minute,
L = length of the conveyor in feet.

The power to operate screw conveyors depends so much upon the specific nature and condition of the fuel and ashes to be handled that the constants in the above equation should be used advisedly. Short-length screw conveyors are commonly used for powdered-coal feeders, and occasionally for sized bulk coal, but conveyors of the screw type are not suitable for handling ashes.

Scraper or Flight Conveyor. This consists of a trough of any desired cross section and a single or double strand of chain carrying flights or scrapers of approximately the same shape as the trough. The flights scrape the material along the trough, discharging at the end through gate-controlled openings in the bottom of the conduit. Three types of flight conveyors are in common use: plain scraper, suspended flights, and roller flight. In the **plain scraper** the flights are suspended from the chain and drag along the bottom of the trough. In the **suspended flight** conveyor the flights are attached to cross bars having wearing-shoes at each end, and do not touch the trough at any point. The **roller flight** differs from the suspended type only in the substitution of rollers for the wearing-shoes. A typical installation of scraper and drag-chain conveyors is illustrated in Fig. 163. The coal conveyor is a single-strand roller flight, 80 ft. in length between centers, driven by a 5-hp. electric motor. It has a capacity of 15 tons of buckwheat coal per hr. The ash conveyor is a single-strand drag-chain with 87 ft. centers on the horizontal run and 6 ft. between vertical centers. The chain operates in an extra heavy cast-iron trough set in a cement trench and is operated by a 5-hp. motor. Flight conveyors are low priced and offer an economical and efficient means of handling coal and ashes in small plants.

Apron conveyors are commonly used for conveying coal from the track hopper to the main conveyor and elevator. The most elementary form consists of flat steel plates attached between two chains and forming a continuous platform or apron. Since the load is carried and not dragged, less power is required than with the scraper type and the maintenance is lower. These carriers are not suitable for elevating material except at an inclination not exceeding 30 deg. End discharge only is possible. Figure 164 shows a typical apron-conveyor installation.

Pan Conveyors and Open-top Conveyors are similar to the apron carriers except that pans or buckets take the place of the flat or corrugated apron plates. These conveyors are used where pans deeper than

those of an apron conveyor are required, as on inclines too flat for elevators and too steep for efficient operation of flight or apron conveyors. Con-

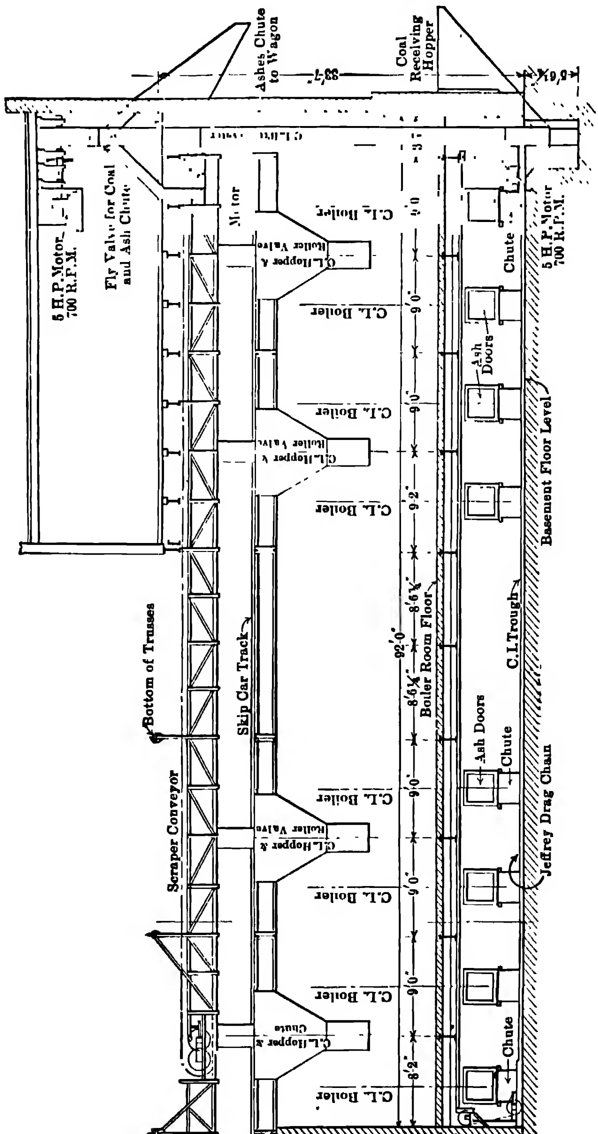


FIG. 163. Typical Scraper and Drag-chain Installation for Handling Coal and Ashes.

veyors of this type are usually run at speeds of 30 ft. to 50 ft. per minute and, when equipped with self-oiling rollers of 6-in. to 8-in. diameter, demand but little power for their operation above theoretical load requirements.

The power required to operate flight, apron, and open-top conveyors may be closely approximated by the following empirical equation:[1]

$$\text{Hp.} = \frac{AWLS}{1000} + \frac{BLT}{1000} + x \tag{62}$$

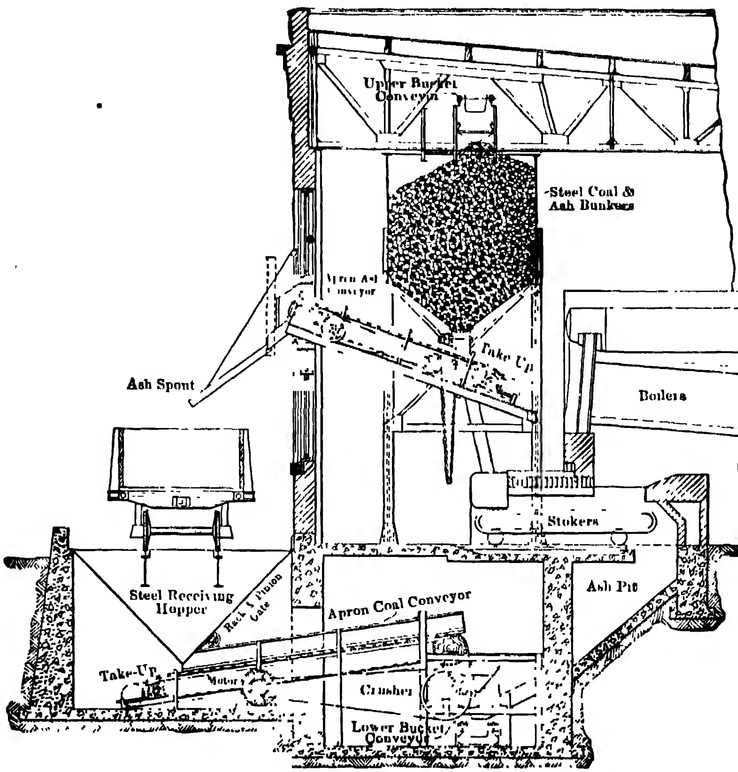


FIG. 164. Typical Apron-conveyor Installation for Handling Coal and Ashes.

in which

Hp. = the hp. required at the conveyor drive shaft,
A, B = constants as in Table 36,
W = weight of conveyor per ft. of run, lb.,
L = distance between centers of head and tail sprockets, ft.,
S = speed of conveyor, ft. per min.,
T = capacity of conveyor, tons (2000 lb.) per hr.,
x = 1, for conveyors up to 100 ft. centers and 2, for longer conveyors.

[1] C. K. Baldwin, The Robins Conveying Belt Co.

If the conveyor is composed of portions on different inclines, compute the power for each section separately and add 10 per cent for each change in direction.

The **V-bucket conveyor** consists of a series of V-shaped buckets rigidly fastened to the conveyor chain. The buckets act essentially as a drag conveyor on horizontal runs, each bucket pushing its half-spilled load

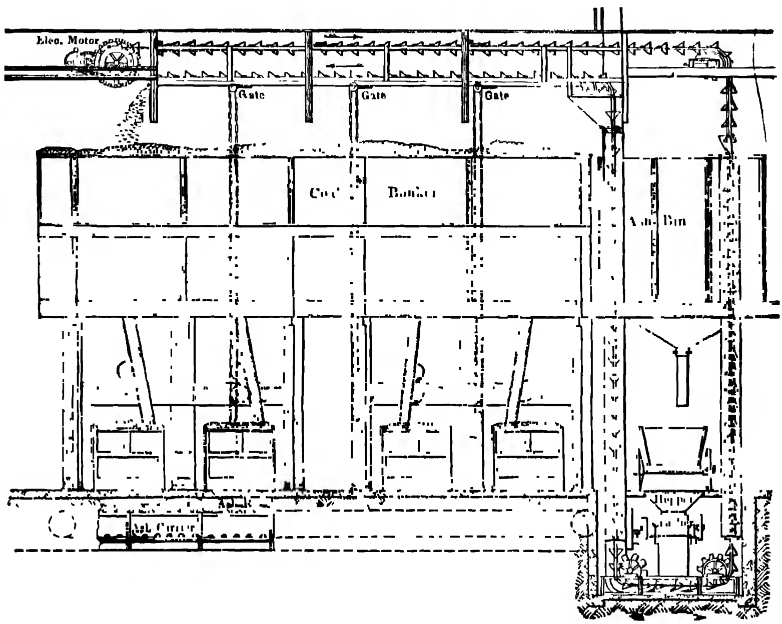


FIG. 165. Typical V-Bucket Installation for Handling Coal and Ashes

ahead of it through a suitable trough. On vertical runs they act as elevators. A typical V-bucket conveyor for handling coal and a pan conveyor for handling ashes are illustrated in Fig. 165. The power requirements may be approximated from the following empirical equations:

$$\text{Hp.} = \frac{AWL'S}{1000} + \frac{BL_1T}{1000} + \frac{TH}{1000} + 1/2\, x' \tag{63}$$

in which

L' = horizontal length of conveyor, ft.,
L_1 = total horizontal length traversed by the loaded bucket, ft.,
H = total vertical traverse, ft.,
x' = number of 90-deg. turns in the conveyor.

Other notations as in equation (62)

TABLE 36

VALUES OF CONSTANTS IN CHAIN-CONVEYOR POWER FORMULAS

Angle of Conveyor with Horizontal Deg.	A.				B. Scraper, Apron and Open Top			B V-Bucket and Pivoted Bucket		
	Sliding Block	3½-in Roller, 3¼-in Pin	6 in Roller, 1¼ in Pin	6-in. Roller, 1½-in. Pin	Anthracite Coal	Bituminous Coal	Ashes	3½-in Roller, ¾ in. Pin	6-in. Roller, 1⅜-in. Pin	6-in. Roller, 1⅜-in. Pin
0	0.030	0.0013	0.0016	0.0050	0.33	0.60	0.51	0.071	0.076	0.083
6	0.030	0.0013	0.0016	0.0050	0.43	0.69	0.63	0.17	0.18	0.19
12	0.030	0.0012	0.0015	0.0049	0.54	0.79	0.73	0.28	0.28	0.29
18	0.029	0.0011	0.0014	0.0018	0.63	0.88	0.82	0.38	0.38	0.39
24	0.028	0.0039	0.0012	0.0016	0.72	0.95	0.90	0.48	0.48	0.49
30	0.026	0.0037	0.0040	0.0013	0.79	1.02	0.97	0.57	0.57	0.58
36	0.025	0.0035	0.0037	0.0040	0.86	1.08	1.03	0.65	0.66	0.66
42	0.023	0.0032	0.0034	0.0037	0.92	1.12	1.07	0.73	0.73	0.74
48	0.020	0.0029	0.0031	0.0033	0.97	1.15	1.11	0.80	0.80	0.81

The **Pivoted Overlapping Bucket** conveyor is perhaps the most popular type of continuous conveyor for handling coal. It consists essentially of a continuous series of buckets pivotally suspended between two endless chains. The buckets at all times maintain their carrying position by gravity whether the chain is horizontal, vertical, or inclined. By means of this system no transfer of material is necessary and discharge may be made at any desired point. Figure 166 gives a diagrammatic arrangement of a pivoted overlapping-bucket conveyor illustrating the principles of a complete coal-handling system, and Fig. 167 illustrates its application to a typical boiler plant.

Referring to Fig. 166, coal is fed to the crusher by the "reciprocating feeder," which is usually placed directly under the track hopper. The feeder consists of a heavy steel plate mounted on rollers and having a reciprocating movement effected by a crank mechanism from the carrier. The amount of coal delivered depends upon the distance the plate moves, and this can be varied by changing the throw of the eccentric. The number of strokes corresponds to the number of buckets. Any size coal can be readily handled. The buckets are made of malleable iron. The capacity of the smallest standard-size carrier is 15–20 tons per hr. at a speed of 30–40 ft. per min., and that of the largest 200–350 tons per hr. at a speed of 50–80 ft. per min. When the distance from track hopper to carrier is so great that the reciprocating feeder is not practicable, an apron or belt cross conveyor is used to supply the crusher with fuel.

The **Hunt Conveyor,** Fig. 168, while usually called a "bucket" conveyor, is in fact a series of cars connected by a chain, each having a body

hung on pivots and kept in an upright position by gravity. The chain is driven by pawls instead of by sprocket wheels. The " buckets " are up-

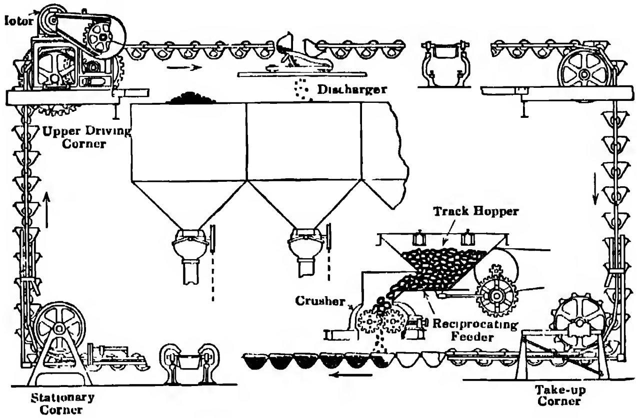


FIG. 166. Diagrammatic Arrangement of a Typical Overlapping, Pivoted-bucket Conveyor and Appurtenances. ("Peck Carrier.")

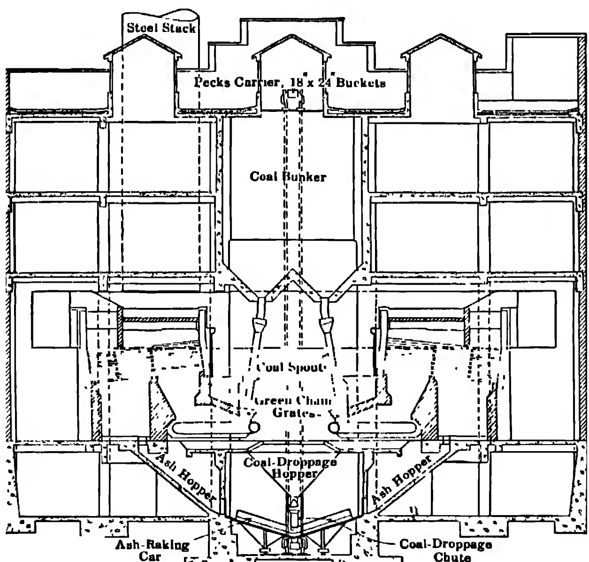


FIG. 167. Pivoted Overlapping-bucket Installation showing Location of Fine Coal and Ash Hopper.

right in all positions of the chain; consequently the chain can be driven in any direction. The change of direction of the chain is accomplished by guiding the carriers over curved tracks. The chain moves slowly,

and the capacity is governed by the size of the buckets. The ordinary size of buckets carry 2 cu. ft. of coal and move at a rate of fifteen buckets a minute, carrying about 40 tons per hour. Two methods of filling the buckets are employed, the " measuring " and the " spout filler." In the former, each bucket is separately filled to a predetermined amount by a suitable "measuring feeder." In the latter, the material is spouted in a continuous stream, necessitating the use of overlapping buckets to prevent spilling of the material.

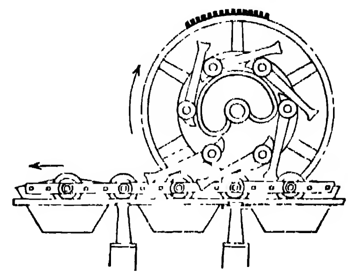


FIG. 168. Driving Mechanism of Hunt Bucket Conveyor.

The power required to operate carrier conveyors of the pivoted-bucket type may be approximated from formula (63), by using the proper value for B as given in Table 36.

Owing to the abrasive nature of ash, the maintenance cost of mechanical conveyors is high. The ash grinds away the connecting pins, and, even with regular renewals, the pins, unless of the enclosed-lubricated type, are apt to wear excessively and cause breakdowns. A breakdown in a long bucket-conveyor system may cause several days' delay before the system can be put into operation again.

Belt Conveyors have a distinctive advantage over most other types of carriers in that they may be driven from any point in their length. The driving machinery is extremely simple; power is applied to one or more pulleys over which the conveyor belt passes. The maximum width of conveyors is limited only by the fiber stress in the belt. Conveyors 1000 ft. from center to center, handling 500 tons per hr., have been successfully operated. Inclinations are limited by the angle of repose of the material. In power plant service they seldom exceed 20 deg.

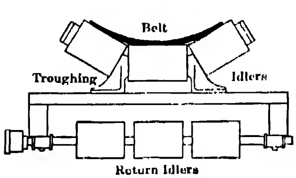


FIG. 169. Arrangement of Pulleys — Belt Conveyor.

The **Robins Belt Conveyor,** Fig. 169, consists essentially of a thick belt of the required width driven by suitable pulleys and carried upon idlers so arranged that the belt becomes trough-shaped in cross section. For heavy duty, five pulleys are employed instead of three, as illustrated, in order that the line of contact may more nearly approach the arc of a circle. The belt is constructed of woven cotton duck, 3–4 ply for 14-in. widths to 8–9 ply for 60-in. widths, covered with a special rubber compound on both sides. An extra covering 1/16 to 1/4-in. thick is used on the carrying side. The rubber is thicker at the middle than at the edges,

since the wear is greatest in a line along the center, but the thickness of the belt is uniform throughout its entire width. The edges are reinforced with extra piles of duck to increase the tensile strength. The idlers are carried by iron or wooden framework, and are spaced from 3 to 6 ft. between centers on the troughing side, according to the width of belt and the weight of the load. On the return side these distances range from 8 to 12 ft. High-speed rotary brushes with interchangeable steel bristles prevent wet, sticky material from clinging to the belt. **Automatic tripping** devices placed at the proper points cause the material to be discharged where it is needed. The trippers consist essentially of two pulleys, one above and slightly in advance of the other, the belt running over the upper and under the lower one, the course of the belt resembling the letter *S*. The material is discharged into chutes on the first downward turn of the belt. The trippers may be movable or fixed, single or in series. Movable trippers are used when it is desired to discharge the load evenly along the entire length, as, for instance, in a continuous row of bins, while fixed trippers are employed where the load is to be discharged at certain and somewhat separated points. The movable trippers are made in two forms, "hand-driven" and "automatic." In the former they are moved from point to point by means of a hand crank. The "automatic" tripper is propelled by the conveying belt through the medium of gearing. It reverses its direction automatically at either end of the run and travels back and forth continuously, distributing its load. It can be stopped, reversed, or made stationary at will.

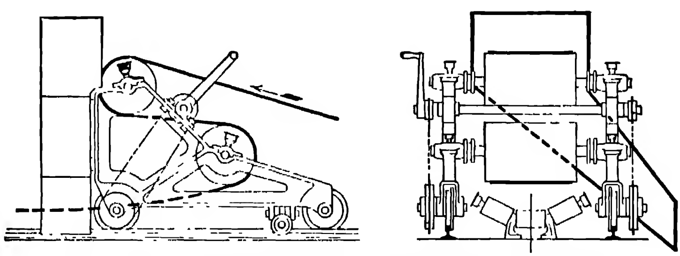


FIG. 170 Hand-propelled Tripper for Belt Conveyor.

The power required to drive belt conveyors may be approximated from the following empirical equations used by the Jeffrey Company.

For level conveyors:

$$\text{Hp.} = (CS + 2.33\ T)\ L \div 33{,}000 \qquad (64)$$

For inclined conveyors:

$$\text{Hp.} = (CS + 2.33\ T)\ L \div 33{,}000 + TH/990 \qquad (65)$$

C = constant as given in Table 37,
S = belt speed ft. per min.,
T = load in tons (2000 lb.) per hr.,
L = length of conveyor between centers, ft.,
H = vertical lift of material.

TABLE 37

POWER REQUIREMENTS FOR BELT CONVEYORS

(Coal and Ashes)

Width of belt	14	16	18	20	24	30	36	42
C	0.75	1.05	1.35	1.70	2.0	2.45	3.55	4.15
Hp required for each movable or fixed tripper	1.0	1.0	1.5	1.5	1.5	2	2	3

Belt-conveyor Operating Data: Power, Oct. 3, 1916, p. 490.
Economics of Conveyor Equipments: Eng. Mag., Nov., 1916, p. 231.

TABLE 38

CAPACITIES OF BELT CONVEYORS

TONS (2000 LB.) OF COAL PER HR. AT VARIOUS BELT SPEEDS

Width of Belt, In.	Belt Speed, Ft. per Min.		Width of Belt, In.	Belt Speed, Ft. per Min.						
	300	350		300	350	400	450	500	550	600
12	34		24	139	162	181				
14	49		30	216	252	288	324			
16	62		36	311	363	414	466	518		
18	80	92	42	423	493	564	635	705	775	
20	96	112	48	552	644	737	830	920	1013	1105

Coal Sorted, Stored and Transported by Belt-conveyor System: Power, Oct. 16, 1923, p. 599.
Using Conveyors in Small Plants: Ir. Td. Rev., Nov. 22, 1923, p. 1428.

122. Skip Hoist. — The skip hoist is one of the oldest, and at the same time the simplest, means of elevating coal or ash, and is finding increasing favor with engineers, particularly in the handling of ash. It consists essentially of a vertical or inclined frame or hoistway, a bucket or car guided by the frame, and a cable for hoisting the bucket. The bucket is so pivoted, with reference to its center of gravity, as to be held in the upright position by its own weight, and the weight of the load. A separate curved guide is located at the dumping point near the top of the hoistway, and engages a roller on each side of the bucket, pulling it into the dumping position. In the modern design the operation is entirely automatic and is

substantially as follows: The skip bucket is filled with coal or ash and the driving motor is put into service from any suitable control point. This automatically starts the hoist, and the bucket is raised to the dumping point. The arrival of the bucket at the dumping point automatically stops the hoist and operates a solenoid brake which holds the bucket at the dumping point for a predetermined length of time. Having remained at the dumping point long enough for the contents to be emptied, the bucket automatically returns to the foot of the hoistway, ready for another load. Maintenance is very low, power is used only when the material is being hoisted, and large hard clinkers are easily handled; but initial cost is comparatively high. This skip hoist offers one of the most satisfactory means of elevating ash, because of its low cost of upkeep.

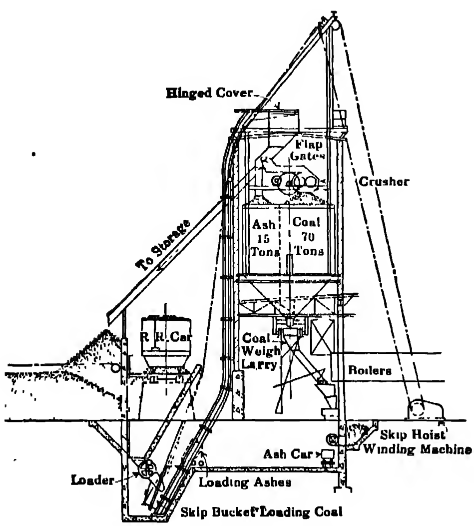


FIG. 171. Typical Skip-hoist Installation.

Storage battery and **gasoline operated trucks, hand cars** and **automatic cable cars** are in use in several plants for handling coal and ashes on horizontal or slightly inclined runs, but they must be either lifted bodily by elevators or the contents must be dumped into suitable conveyors for vertical lifts.

123. Monorail: Telpherage. — The **telpher** is a form of hoist which lifts and transfers the load on a single rail or track from one point to another. Both hoisting and travel may be accomplished by either hand or power. Where electric power is used, the hoist and carriage are made in two forms: one, in which both the hoisting and travel are controlled from the ground and it is necessary for the operator to walk with the carriage; the other, in which the operator rides in the car and manipulates the control from the carriage. Figure 172 illustrates a very simple and economical method of handling coal and ashes as installed by the Jeffrey Mfg. Co. at the power plant of the Scioto Traction Co., embodying the telpher systems. If the coal car is of the dump type, the contents are discharged directly into the coal pit, from which the coal is removed by grab bucket and transferred either to the overhead bunker or to the storage pile. If the coal car is of the gondola type, the coal is removed directly from the car by the grab bucket. The bucket is hoisted and carried on the trolley into the

building over the screen hoppers, where it discharges its contents; the finer particles fall directly into the bunker and the larger lumps are automatically delivered to the crusher. The grab bucket will take about 98 per cent of the coal in the car, leaving only 2 per cent to be removed by hand. Coal is fed to the stokers by means of a traveling electric hopper which receives its supply from the overhead bunkers. The present capacity of the plant is 50 tons per hr., taken from the car or pit to stock pile.

Pneumatic and Hydraulic Systems. See paragraph 124.

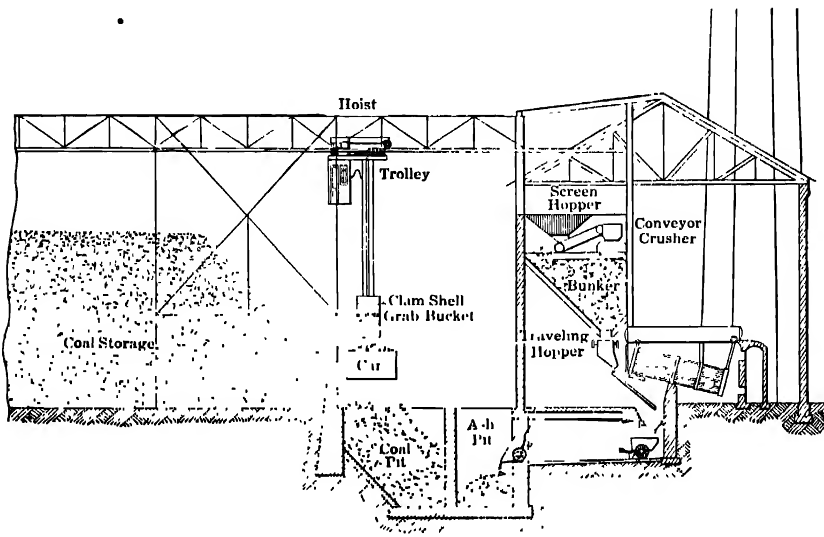


FIG. 172. Typical Telpherage System.

124. Ash-handling Systems. — While many of the various systems of handling coal can be applied to the handling of ash and other fuel refuse, it must be remembered that the latter may be dripping wet, red hot, dry and dusty, soft, hard, granular, or in the shape of large clinkers; and that the abrasive action of all ash, dry or wet, and the corrosive action of wet ash seriously affect the life and maintenance of rubbing surfaces and sheet-metal parts. For this reason it is good practice, particularly in large stations, to handle the fuel and refuse with independent systems and with as little machinery as practicable.

Gravity Systems. One of the simplest and most efficient means of removing ash is to dump it directly from hopper ashpits into railroad cars, without the use of any machinery other than that required to open and close ash gates. This system is applicable only where the firing aisle is designed for a considerable height above ground level. Such an installation is shown in Fig. 175. The only maintenance required is upkeep of

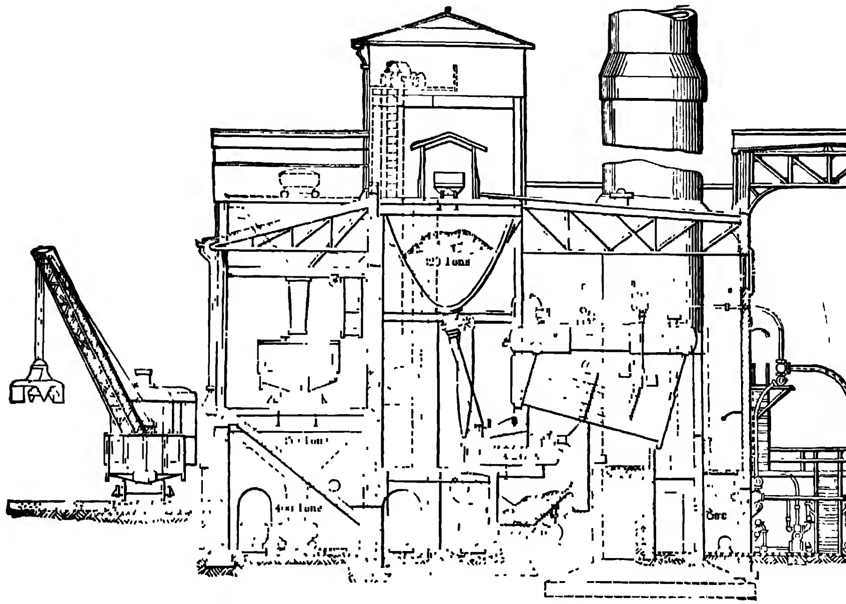


FIG. 173. Typical Installation — Portable Crane. Grab Bucket and Bucket Conveyor.

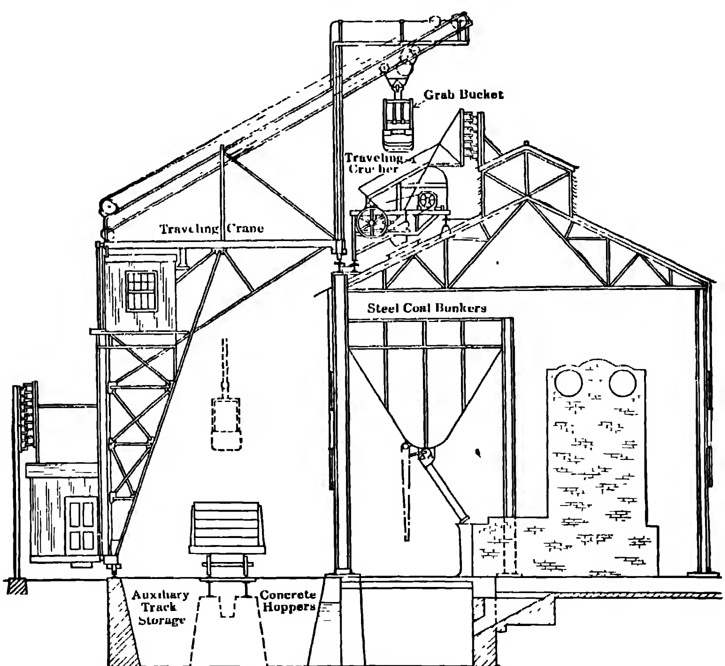


FIG. 174. Grab-bucket Installation, American Rolling Mills Co., Middletown, Ohio.

ashpit linings and dumping doors. Caterpillar tractors, motor trucks, industrial cars, or wagons are used where railroad tracks do not enter the plant.

The Hydraulic Conveyor or Sluice. This system is another example of

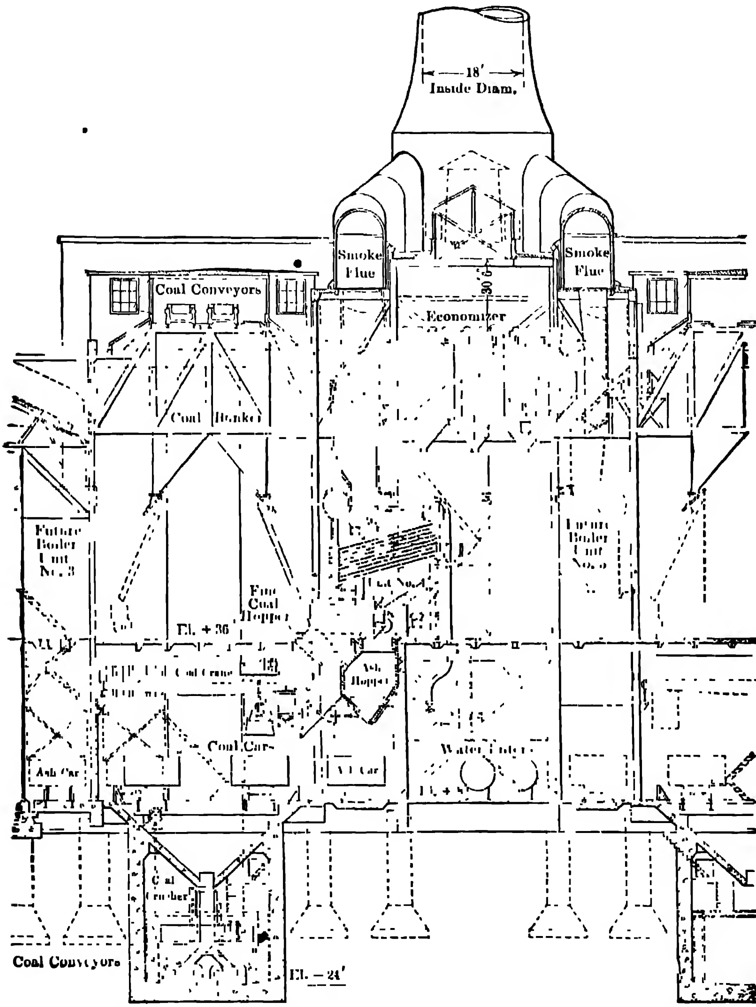


FIG. 175. Coal and Ash Handling System at "Northwest" Station.

horizontal conveyance which has many good features. In this system a stream of high-velocity water flows in flumes or open conduits underneath the ashpit and carries the ash to waste or to a sump from which it is reclaimed by suitable means. With continuous-dumping stokers, such as

chain-grates or underfeeds equipped with clinker grinders, the ash may be discharged directly into the running stream; but with dumping stokers or any firing system where large clinkers are to be expected, the refuse is dumped on to a **grizzly** or massive bars on which the clinkers may be broken. Where ash is discharged intermittently, or where large hoppers are provided with continuous-discharge stokers, the water need be running only while dumping is in progress. With natural draft, the ashpit should be sealed so as to prevent excess air from passing up the stoker dumps. A typical installation of this class of conveyor is at the Hell Gate plant, Fig. 176.

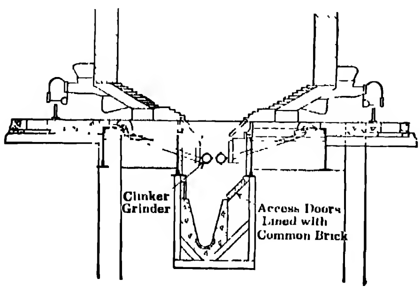


FIG. 176. Water Conveyor at "Hell Gate" Station.

Open flumes of concrete, with a bottom lining of vitrified earthen drain tiles, are installed below each line of boilers and discharge into a collecting cross flume which runs along the boiler wall near the turbine room. The cross flumes lead into a closed conduit which in turn discharges into a pit near the river. The ash is recovered from this pit by a locomotive-type grab bucket and discharged into scows, and the water overflows into the river. The water supply is taken from the condenser circulating discharge tunnel, pumped against a head of 75 ft. by three 12-in. centrifugal pumps connected to 150 hp. motors, and discharged into the flumes through a series of nozzles. At the Lacomb Station of the Denver Gas & Electric Light Co., the water is passed through a screen and recirculated instead of being discharged to waste.

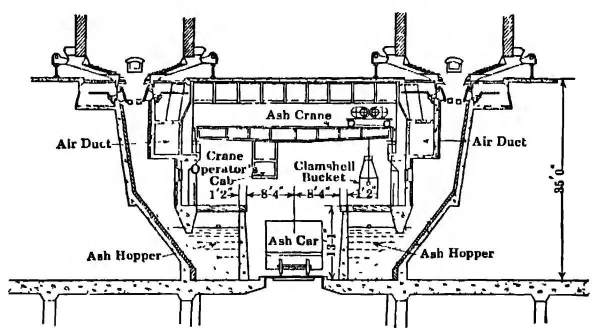


FIG. 177. Ash Handling System at "Springdale" Station.

Submerged Cross-bar Conveyor. The scraper conveyor has been a favorite with engineers for years, but it is only within the last few years that this system of conveyance has been applied to water-filled troughs. In the latest installations, both the upper and the lower chains run under

the surface of the water (see Fig. 178). Ordinarily no crusher is necessary, as the space between chains and cross bars is sufficient to allow loose clinkers to fall through the upper runs of the chains into the water; and the hot clinkers are disintegrated by the action of the water. With stokers of the continuous discharge type, no ashpit is necessary and a seal is effected by dipping the ash spouts below the water level of the trough.

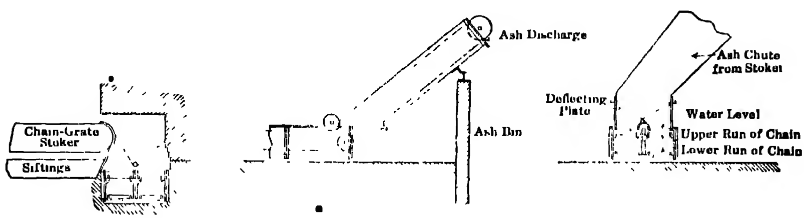


FIG. 178. Cross-bar Conveyor and Water-seal Chain.

There is no flow of water, and it is only necessary to add enough makeup water to take care of that absorbed by the ash. Excess water is removed by elevating the discharge end of the trough so that the surplus will gravitate back to the trench.

Pneumatic Systems. When air is passed through a pipe at sufficiently high velocity, it is capable of carrying dry solid material of considerable size along with it. The high velocity may be established by forcing air

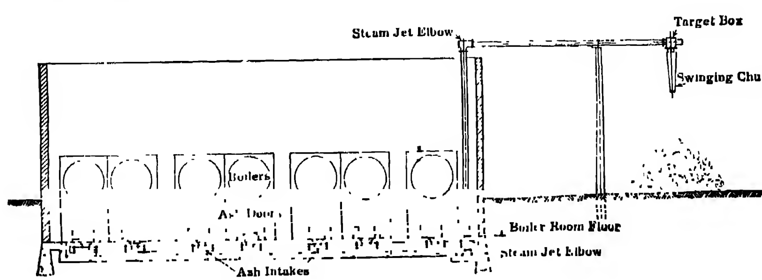


FIG. 179. A Typical Vacuum Ash-handling System.

through the pipe under pressure greater than atmospheric, or by creating a partial vacuum in the pipe. The pressure system is commonly used in conveying powdered fuel from one point to another, and the vacuum system in removing refuse from ashpits and combustion ash from uptakes, economizers, breechings, etc. The vacuum system has also been used for conveying bulk coal but only to a very limited extent.

The vacuum system consists primarily of a line of pipes into which the ashes are fed and through which they are carried to a discharge point by

the air current. Ash or soot intake fittings, which can be closed when not in use, are installed at suitable points in the pipe line, and an air intake is provided at the extreme end of the line. In some of the older plants and in a few modern installations where conditions are favorable, the vacuum is created in an air-tight storage tank at the discharge end of the pipe line,

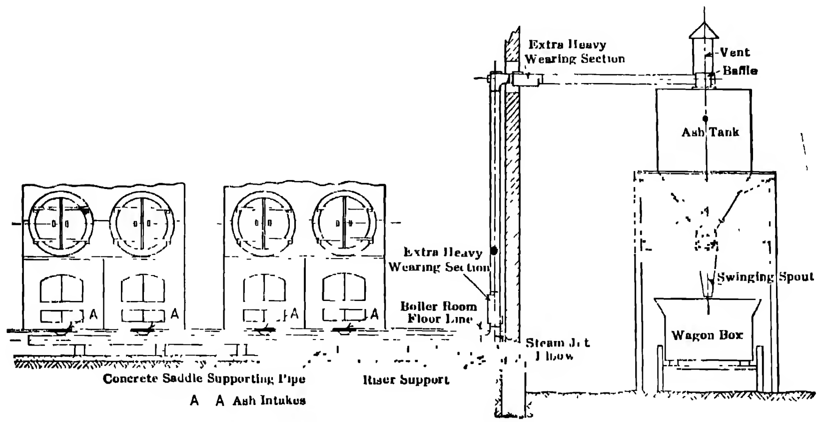


FIG. 180. A Typical Vacuum Ash-handling System.

the entire system being under a partial vacuum. This method is used so little in steam power plant practice that no attempt will be made to describe it.

Figures 179 and 180 show applications of the modern vacuum system for removing refuse from the ashpit. It will be seen that the suction is created by one or more steam jets placed between the pipe inlet and discharge tank. Only that portion of the pipe line between the intake openings and the jet is under suction, the portion beyond being usually under pressure. The nozzles producing the jet are of monel metal or hard brass and are of the divergent type (angle 8 to 10 deg.). They are inserted in special fittings, as shown in Figs. 181–3. These fittings may be of the straight or angle type, depending upon whether they are to be inserted in the straight pipe run or in elbows or bends. Only one nozzle, Fig. 181, is used in the **angle,** while two nozzles, Fig. 182, are ordinarily inserted in the **straight line** fitting. A single nozzle in an offset fitting, as in Fig. 183, is used by one manufacturer in place of twin nozzles in a straight run. The effective suction distance of a single-nozzle fitting is limited to approximately 50–80 ft. The effective discharge distance is much

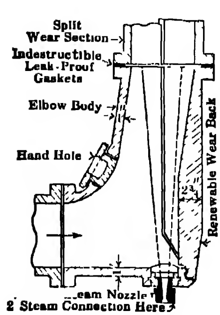


FIG. 181. Typical 90-degree Jet Elbow.

greater than its suction distance; therefore, in the average installation, no additional steam units are necessary in the discharge line. For long runs of pipe, two or more nozzle fittings, or **boosters,** are required. Ashes can be moved economically by air conveyors through a horizontal distance of about 500 ft. and to a vertical elevation of about 100 ft. Pipe sizes range from 6 to 9 in. in internal diameter. The maximum capacity of a 6-in. conveyor is approximately 4 tons of ash per hr., that of the 8-in., 6 to 9 tons, and that of a 9-in., 10 to 15 tons. Sizes above 9 in. are not practical because of the amount of steam required to produce the necessary suction. Ashes should not be wet or quenched when fed to an air conveyor, and, of course, must be small enough in size to enter the inlet openings. Steam-jet vacuum conveyors use a large quantity of steam while running; but since they remove the ash very rapidly, the cost of steam per ton of ash removed is comparatively small, provided the nozzles are correctly proportioned and the ash is supplied at a rate near the maximum capacity of the line. Steam-jet conveyors cannot be used with low-pressure plants, since it has been found by experience that a minimum pressure of 60 lb. gage is required at the nozzle for successful operation. One lb. of steam will move from 6 to 16 lb. of ash, depending upon the character of the ash, the initial conditions of the steam, design of nozzle and piping, and the rate of feeding the ashes into the pipe. Steam-jet vacuum conveyors are usually lower in first cost than a mechanical conveyor system of equal capacity, take up very little space, can be installed in awkward positions, result in cleaner basements or firing floors, and ordinarily require little attention. Because of the abrasive action of the ash, moving at high velocities, considerable erosion of the pipe and fittings takes place. The wear is greatest at the elbows where the direction of flow is changed, and it is customary to provide **wearing blocks** which can be readily replaced. A **baffle box** is usually installed at the top of the ash bin, to break the force of the ashes just before they drop into the storage tank. A water spray should be installed in connection with all conveyors discharging into the open or into a baffle box.

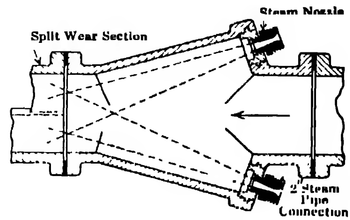


FIG. 182. Twin-jet "Booster" or Straight-run Nozzle Fitting.

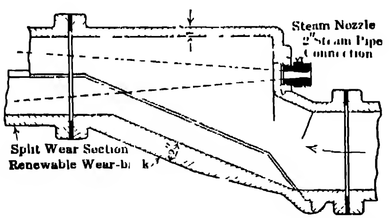


FIG. 183. Single-jet "Booster" (United Conveyor Corporation).

Among the popular designs of vacuum ash-handling systems may be mentioned those manufactured by the Conveyors Corporation of America, United Conveyors Corporation, Brady Conveyor Co., and M. H. Detrick Co.

Ash Handling: by John Hunter and Alfred Cotton, Trans. A.S.M.E., Vol. 44, 1922, p. 687.

125. Typical Installations in Modern Central Stations. — Figure 184 shows a section through part of the boiler room of the Calumet Station of the Commonwealth Edison Co., illustrating an application of several types of conveyors to a modern central station. Coal is delivered to the plant in railroad cars, which enter the boiler basement on tracks directly underneath the firing aisle. A traveling crane equipped with a **clam-shell bucket** is used to remove the coal from the cars[1] and to deliver it either to the storage bin adjacent to the track or to a traveling hopper. This hopper is equipped with an oscillating feeder by means of which the fuel is fed to a belt conveyor. The latter is arranged so as to deliver the coal directly to a pivoted-bucket conveyor (installed in duplicate) for immediate delivery to the overhead bunkers in the boiler room, or to a Bradford coal breaker, depending upon whether the coal is in the shape of screenings or whether it requires breaking. The coal from the breaker is carried by two belt conveyors, installed at right angles to each other, to a Robins double-roll crusher. The fuel passing through this crusher is finally delivered to the pivoted-bucket conveyors previously mentioned. The crusher with its associated conveyors is for emergency in case the breaker is out of commission. Each of the elements of the conveying system is motor-driven through enclosed gear speed reducers. Ashes are dumped directly into railroad cars through pneumatically operated gates. The ash hoppers are provided with a sprinkling system for quenching the hot cinders and wetting the ash.

Figure 185 shows a section through the boiler room of the Windsor Station of the American Gas & Electric Co. and the West Penn. Power Co., illustrating a simple and efficient system of coal and ash handling. Coal is dumped from the railroad cars into a concrete pit which runs the entire length of the boiler room beneath the firing aisle. From this pit the fuel is lifted in a 3 cu. yd. **grab bucket** operated from an overhead crane. After being weighed by a device on the crane, it is discharged into the individual boiler hoppers. From the hopper, the coal gravitates through down spouts to the stoker hopper. Ash is stored in pits and the accumulation dropped into transfer cars.

In the South Meadow Station of the Hartford Electric Light Co., Fig. 186, coal is dumped into covered track hoppers. From the track

[1] Revolving car dumper now used for this purpose.

hopper, the fuel is discharged upon an apron conveyor, which delivers it to two roll crushers. Chutes and by-passes permit discharge from either

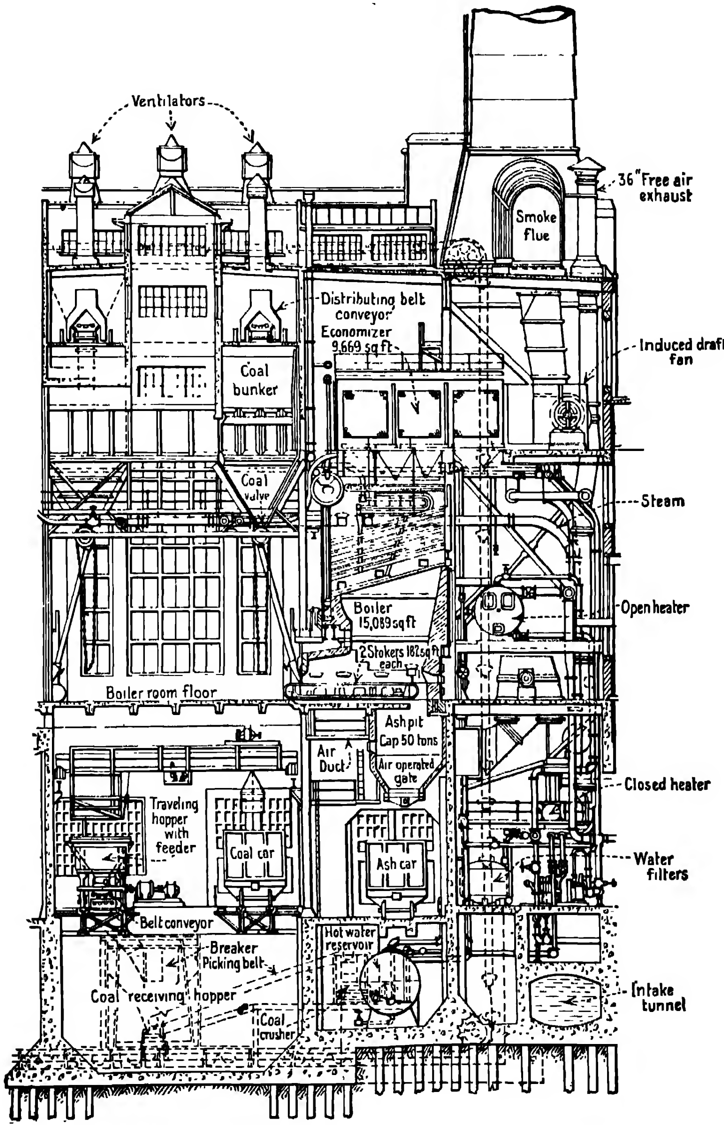


FIG. 184. Coal and Ash-handling System at "Calumet" Station.

conveyor to either crusher, also from either crusher to either of the two bucket elevator conveyors. These elevators deliver the coal to the central bunker. Drives of conveyors, crushers, and electric elevators

are induction motors with controls so interlocked that starting and stopping can be done only in proper sequence.

126. Powdered-fuel Conveying Systems. — Powdered fuel is usually moved from mill to storage, from storage to burner, or directly from mill to burner, by combinations of screw feeders and pneumatic conveyors. Conditions occasionally arise, however, where it is more economical to convey the powdered product in bulk by tank cars, barges, etc. Among

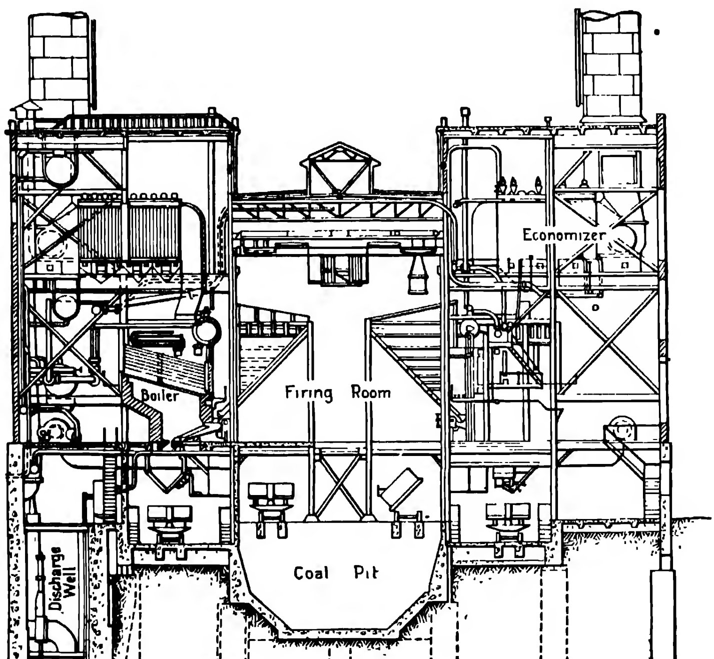


FIG. 185. Coal and Ash-handling System at "Windsor" Station.

the popular powdered-fuel handling systems may be mentioned the **Lopulco, Rayco, Grindle, Quigley** (now incorporated with the Fuller-Lehigh Co.), **Fuller-Kinyon, Holbeck** and that of the **Ground Coal Corporation.**

In the **Quigley** system, the mixture of powdered fuel and air passes from the pulverizer through a special separator, where the oversized particles are removed and returned to the mill. The finished product and the air entrainment are withdrawn from the top of this separator by an exhaust fan and discharged into an overhead vented-cyclone dust collector where the air and dust are separated. The pulverized fuel gravitates from the collector to the powdered-fuel hopper, and the air is returned to the bottom of the separator. From the hopper the fuel is fed by gravity into a blow-

ing tank, until the desired quantity has been deposited. A dust-tight inlet valve on top of the tank is then closed and compressed air is admitted. When the desired pressure is reached, a discharge valve is opened and the fuel is conveyed to the hopper over the furnace. The fuel is discharged

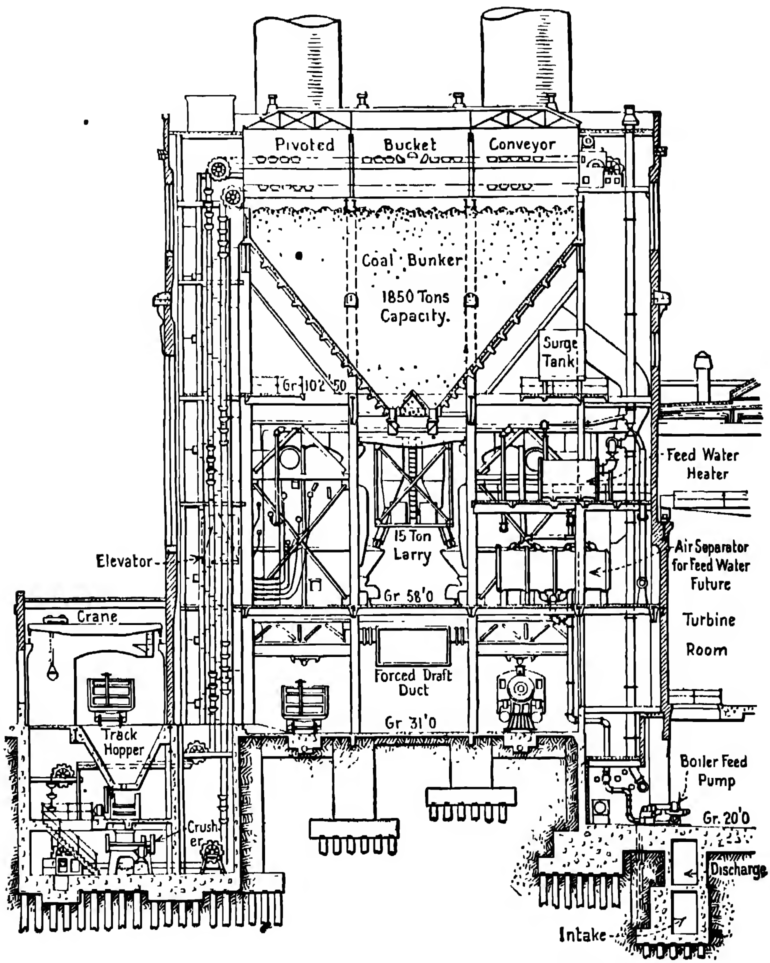


FIG. 186. Coal and Ash-handling System at "South Meadow" Station.

through the distributing pipe in slugs, a sort of pulsometer action taking place between the air and fuel. Air is compressed to 100 lb. per sq. in. and then expanded to the pressure required to start the fuel. The air requirements are about 1 cu. ft. of free air for each 1.5 to 2.0 lb. of fuel transferred. With this system the fuel may be moved at a rate of 50 tons

per hr. through a 4-in. pipe for various distances up to 4000 ft. **Blowing tanks** are supplied singly or in as many units as conditions dictate. A notable installation of the Quigley system is in the Cahokia Station of the Union Electric Light & Power Co., St. Louis, Mo. A complete layout of a Quigley powdered-coal plant is shown diagrammatically in Fig. 187.

In the **Holbeck** low-pressure distributing system, the pulverized fuel is delivered from the vacuum separator of the mill through an exhauster to the cyclone separator. The air returns to the pulverizer through a return pipe, while the fuel drops into a central bin and is withdrawn from

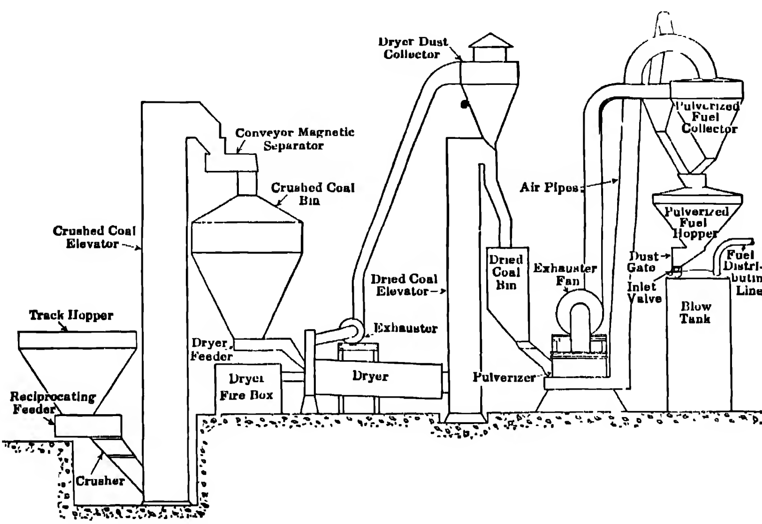


FIG. 187. Typical Powdered-fuel Plant — Quigley System.

the bottom of the latter by a feed screw. This screw delivers the fuel into the suction side of a high-pressure blower. From this point on, the fuel is blown through the distributing mains directly to the burners. The remaining air and fuel, which are not used at the furnaces, are returned through an auxiliary line to the collector and are separated. The fuel returns to the bin and the air to the suction side of the blower. The return of the surplus air and fuel permits the maintenance of sufficient velocity in the distributing line to keep the fuel in suspension irrespective of the number of burners in operation. About 25 per cent of the air required for combustion is used in the distributing main, a ratio of approximately 50 cu. ft. of free air per lb. of fuel.

Figure 188 gives the general details of the **Fuller-Kinyon** system. Powdered fuel is fed from the bottom of the storage hopper into the "pump," which is essentially a worm or screw revolving in a closed chamber. The

worm moves the fuel to the end of this chamber, where it is aerated by a small volume of compressed air. The air-fuel mixture is forced from this point through a reducing nozzle to the various bins. Beyond the nozzle is a valve which can be closed while the air line is being blown out with air. Each bin is provided with a vent pipe, but no cyclone is considered necessary owing to the small amount of air used. Air requirements are approximately 1 cu. ft. of free air per lb. of powdered fuel. Air pressures vary with the distance and range from 5 lb. for a horizontal distance of 100 ft. to 50 lb. for a horizontal distance of 3000 ft. Power consumption

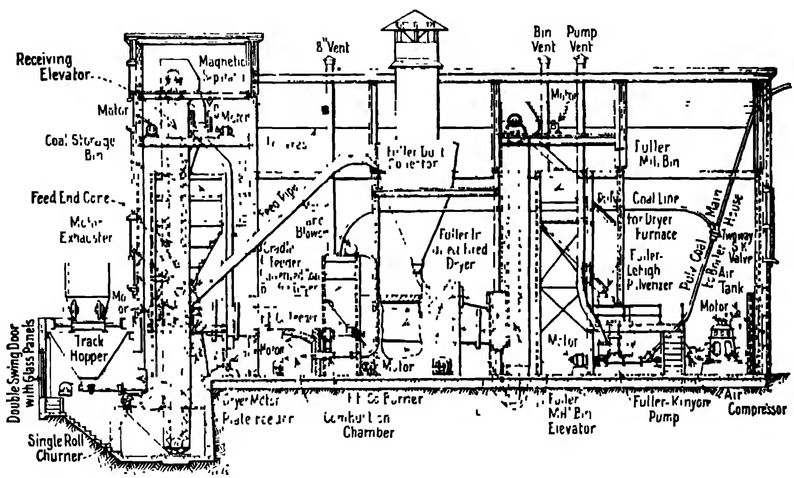


FIG. 188. Typical Powdered-fuel Plant — Fuller-Kinyon System.

of compressor and pump varies from 1.2 to 2.0 hp-hr. per ton of fuel conveyed. (See, *Power*, Aug. 5, 1924, p. 215.)

A notable installation of the Fuller-Kinyon system is at the Lakeside Station, St. Francis, Wisconsin, of the Milwaukee Electric Railway & Light Co.

Preparation, Transportation and Combustion of Powdered Coal: Bureau of Mines, Bul. No. 217, 1923; Power, June 3, 1924, p. 900

Test of a Powdered Coal Plant: H. Kreisenger, U. S. Bureau of Mines, Tech. Paper 316, 1923.

127. Fuel-oil Feeding Systems. — Oil may be transferred from the supply tank to the burner by (1) gravity feed, (2) column gravity feed, (3) compressed air, and (4) steam or motor-driven oil pumps. All of these systems may be found in present-day operation, but by far the great majority in steam power plant practice are of the oil-pump class; for this reason, no attempt will be made to describe any but the pumping

systems. All oil-feeding systems must be installed in accordance with Underwriters' requirements and community ordinances, except, of course, where there are no restrictions and fire insurance is not desired.

Figure 189 gives a diagrammatic arrangement of the equipment and piping in a typical " oil pump " system, illustrating current practice with burners of the steam-atomizing type. Steam-actuated oil pumps, installed in duplicate, draw the fuel from the service tank and deliver it under pressure to the burners. The oil supply to the pump must be of sufficiently low viscosity to flow freely. With many of the low-Baumé

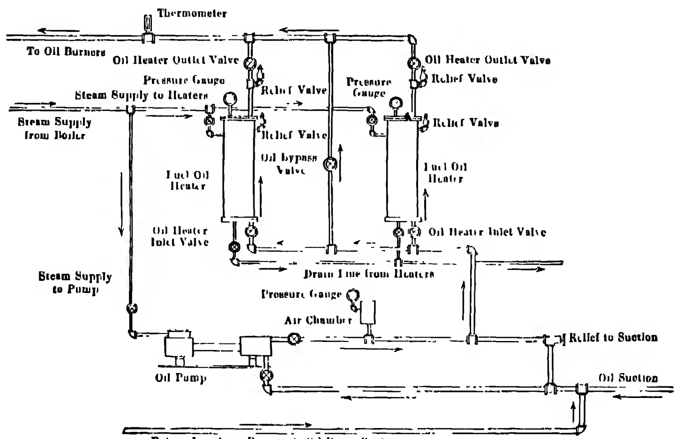


FIG. 189. Diagrammatic Arrangement of Piping for Fuel-oil System (Steam Burner).

fuel oils, this necessitates preheating to between 90 and 110 deg. fahr. The piping is cross-connected so that repairs can be made without interrupting the service. The oil is forced by the pump through a heater receiving its heat from the pump exhaust. With the steam-atomizer type of burner, oil temperatures at the burners above 160 deg. fahr. are seldom necessary. Therefore, the pump exhaust has sufficient temperature to effect the necessary heating, provided, of course, the amount of steam is ample. The heating of the oil should not exceed the vaporizing point under the existing oil pressure, otherwise a sputtering flame may result. Strainers of the duplicate type are placed between the supply tank and pump suction, and in the oil-feed line between the pump and burners The relief valve between the pump and burners is set at a definite maximum oil pressure so as to prevent excessive pressure. All oil piping is installed so that it can be drained back to the storage tank by gravity in case of necessity. In small plants, the oil and steam pressures are usually regulated by hand at each individual burner. In many plants the oil and atomizing steam pressures and the air supply are automatically

controlled. A popular automatic control, used principally on the Pacific Coast, is the **Moore-Patent** automatic fuel-oil regulating system. This system controls the supply of oil to all burners, the supply of the atomizing agent to all burners, and the supply of air for combustion, for any number of burners, all from a central point. In this system all individual burner valves, both steam and oil, are opened wide or nearly so, and all burners are operated under full pressure in the respective mains. In the larger plants, all dampers are connected to a common rocker shaft and move simultaneously. A slight variation in the steam pressure on the boilers, due to any variation in the demand for steam, is the primary means of control for a steam regulator or governor which varies the oil pressure at the oil pumps and in the oil main. The supply of steam to the burners is controlled by regulating the pressure in a separate low-pressure main common to all burners, the pressure in the steam main bearing a certain predetermined relationship to the pressure in the oil main and being controlled by a ratio regulator. By means of a specially constructed diaphragm regulator, the opening of the boiler dampers is made to increase or decrease with a corresponding variation of pressure in the oil main. For a description of this apparatus, consult Trans. A.S.M.E., Vol. 30, 1909, p. 804. The relation between boiler rating and oil and steam pressures for a special case is given in Fig. 190

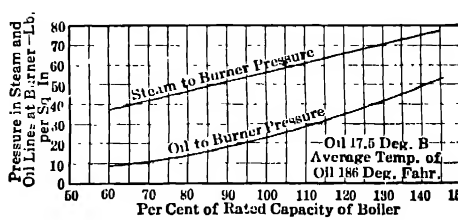


FIG. 190. Relation Between Oil and Steam Pressure — Steam Atomizers, Automatic Control.

TABLE 39

TEMPERATURE OF FUEL OILS FOR MECHANICAL ATOMIZATION

(Peabody Engineering Corporation)

Specific Gravity of Oils Deg. Baumé	Temperature to Which Oil Should be Heated Deg. Fahr.	Specific Gravity of Oils Deg. Baumé	Temperature to Which Oil Should be Heated Deg. Fahr.
10.5	330	18	185
11	315	20	160
12	290	22	140
13	270	24	125
14	250	26	110
15	230	28	95
16	210	30	85

The oil-feeding system for mechanical burners differs from that of steam burners only in the elimination of the steam line to burners and in the use of higher oil pressures and temperatures. The pressure depends upon the viscosity of the oil and the rate of combustion, but ranges approximately from 75 to 250 lb. per sq. in. The viscosity of liquid fuel is not strictly a function of specific gravity. Water-gas tar, for example, while very heavy (in fact, heavier than water) is at the same time very fluid and requires no heating for use in mechanical atomizers. For average fuel oils, the temperatures are approximately as in Table 39. The oil pumps may be steam or motor driven, but should be of the rotary type in order to avoid pulsations. The temperature of the oil is ordinarily above that of the pump exhaust, and the heaters are therefore of the live-steam type. The full-line oil pressure should always be maintained at the tip of a mechanical atomizer, the individual burner valves always being left open. The variation in load may be handled either by cutting in or cutting out the individual burners, or preferably by varying the oil pressures at the pump, or through the agency of a master valve leading to the burners. Mechanical oil burners are readily controlled automatically. In the **Merit Automatic Oil System** the oil pressure on the discharge side of the pump is governed through a steam-actuated **master controller set,** directly connected to the main steam header. This "master controller set," located near the pumps, operates a damper interlocking device on the front of each battery of boilers and controls the fires and dampers in three predetermined steps. Separate and independent adjustments are provided for the pressure of oil to burner, for flue damper and speed of fans (when used). As the mechanism controlling the fire is interlocked with that governing the draft, a change in the firing cannot be made until the dampers are in proper position.

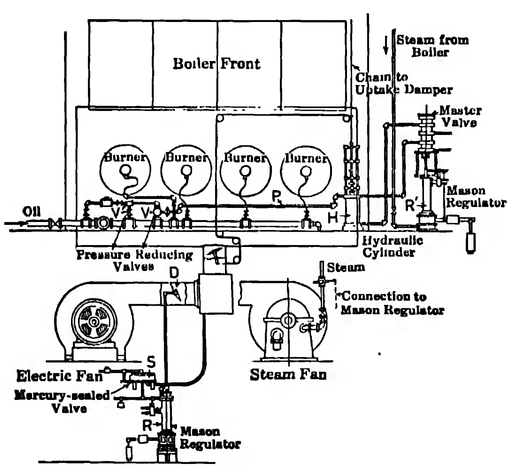


FIG. 191. Diagram of Automatic Oil-stoking Control Narragansett Elec. L't'g. Co.

Figure 191 gives a diagrammatic arrangement of the automatic oil-stoking system at the power plant of the Narragansett Electric Lighting Co. The mercury-sealed bells, actuated by duct pressures, control the

forced-draft fan through the opening of damper D, Mason-type regulator R if the fan is motor-driven, or through regulator C and a chronometer valve in the steam line if the fan is turbine-driven. M is the master valve operated by variation in main steam pressure by means of the Mason-type regulator R; H is the hydraulic cylinder and piston which operate the forced-draft damper, or, if it is used with steam-atomizing burners, the stack damper. Water pressure through pipe P actuates the diaphragm valve; V and V' are pressure-reducing valves through which the oil passes for a low fire.

The **Balanced-Draft, Hagan** and **Ruggles-Klingeman** combustion-control systems are also designed to meet the requirements of fuel-oil furnaces.

Burning Liquid Fuels: Report of Prime Movers Committee, N.E.L.A., T3 1922, p. 254; Part B, 1923, p. 297.

Fuel Oil Unloading Apparatus: Power, Dec. 7, 1920, p. 890.

Pipe-line Transmission of Crude Oil: Power Plant Engrg., Dec. 1, 1919, p. 1039.

The Merit Automatic Oil-stoking System: Power, Oct. 4, 1921, p. 531.

Efficient Heating of Oil Fuel: Ind. Management, Vol. 66, July, 1923, p. 36.

128. Coal-weigh Larries — Coal Valves. — The weighing of fuel is just as important to the economical operation of the boiler plant as is the weighing of raw material to that of an industrial plant. It is surprising how many boiler plants, large and small, make no attempt whatever to weigh the fuel after it has been delivered to the plant, keeping no records for determining the fuel consumption but the fuel tickets furnished by the distributors. While, with certain types of stokers, it is possible to closely approximate the rate at which fuel is being fed to the furnace by the speed of the grate or the number of strokes of the feeding ram, the majority of plants depend upon intermittent weighing of the supply as it is fed to overhead bunkers or to the individual stoker magazines. The most popular method in the large modern boiler plant is to weigh the fuel in a traveling hopper scale called a **larry**. The tracks

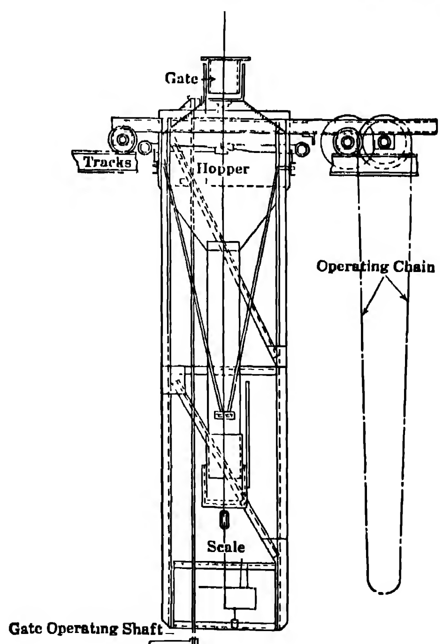


FIG. 192. Coal-weigh Larry — Hand-propelled.

for the larry run under the down spouts of the overhead bins or storage hoppers, and above or on the boiler-room floor, depending upon whether the track is of the suspended type or is laid on the floor. The larry may be hand-operated and hand-propelled, motor-propelled and hand-controlled, or motor-propelled and motor-controlled, depending upon the size. Larries are constructed in various sizes ranging in capacity from 1/2 to 25 tons.

Stationary weighing hoppers may be installed above each stoker magazine, but the first cost is apt to be prohibitive. A typical installation is shown in Fig. 193. The bottoms of the overhead coal bunkers lead into the small hoppers *A*, *A*. The operation of any single weighing hopper is as follows: Coal is fed from the overhead bunkers to weighing hopper *H* by means of valve *V*. The weight of coal in the weighing hopper is transmitted by a system of levers and knife edges to the enclosed scale beam *I* and noted in the usual way. The weighed charge of coal is then admitted to the down spout *S* by means of valves similar to those at *V*. Weighing hoppers are sometimes made automatic; that is, the opening and closing of valves, feeding of coal, and recording of weight are automatically performed by the weight of the coal itself. The scale is set for discharges of a certain weight and continues to discharge this amount automatically. In the few plants which are equipped with automatic weighing hoppers, the capacity of the hopper is approximately 100 lb. per discharge.

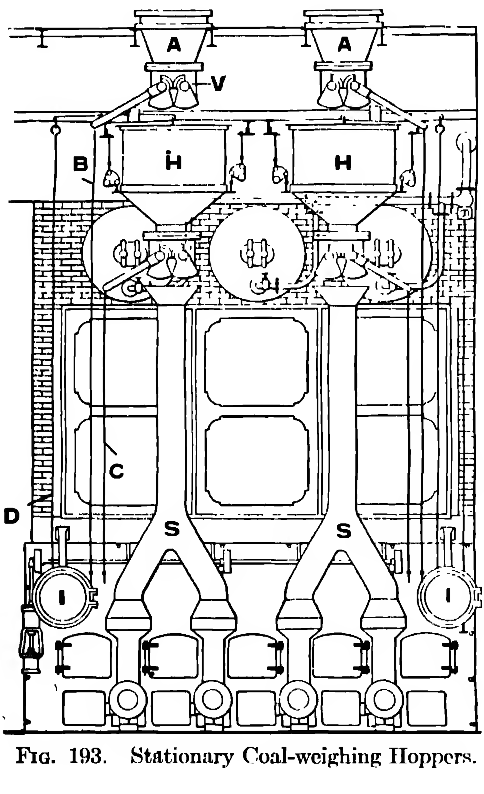


FIG. 193. Stationary Coal-weighing Hoppers.

Figures 194–5 illustrate the principles of a few types of coal valves. They may be conveniently grouped into two classes according to the location of the coal pocket: (1) those drawing the coal from overhead bunkers and, (2) those drawing from the side of a bin. In the first class

come the **simple slide valve** and the **simplex** and **duplex rotating valve.** In the latter class are the **flap valve** and the **rotating valve.** They are made in various sizes and designs, but those illustrated are examples of the more common types. The simple slide valve is applicable only to small-size coal and to small spouts, since coarse or lump coal may get in the way and prevent proper closing. The simplex valve consists of a rotating jaw actuated by a lever. There are no rubbing surfaces, and the jaws cut through the material without jamming. The duplex valve consists of two rotating jaws connected to a common actuating lever. The jaws move simultaneously, so that even a partially open valve delivers the coal centrally. When the valve closes, the flow is gradually stopped by the decreasing width of the opening and there is but little resistance to the movement of the jaws. The largest valve can easily be operated by hand.

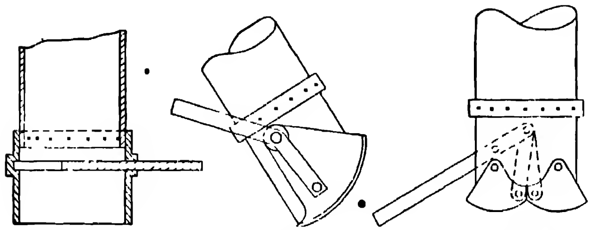


Fig. 194. Typical Coal Valves for Vertical Discharge.

The flap valve is the simplest form for drawing coal from a side bin. It consists merely of an iron flap hinged to the bottom of the chute. The valve is lowered to let the coal run over its top and is raised to stop the flow. It cannot be clogged or jammed in closing. The flap is raised and lowered by a simple lever. For very large bins, where the valves are to be opened and closed frequently, the "Seaton" valve is usually preferred. This valve consists of two jaws, EE' and TT', pivoted to suitable framework at O and actuated by lever A. The valve is shown fully closed. Raising lever A causes the cut-off blade EE' to rotate about O and permits the coal to flow through the space between the edge of the jaw E and the end of the chute. The cut-off blade does not reach a stop; hence there is no possibility of a lump of coal getting in the way and preventing the prompt closing of the valve.

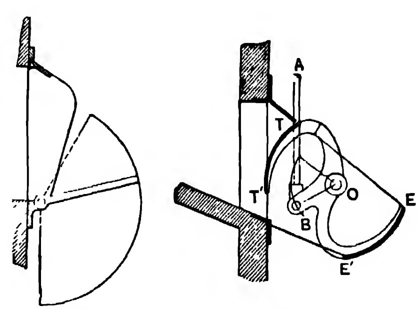


Fig. 195. Typical Coal Valves for Side Discharge.

Coal and Ash Handling Systems:

Waukegan Station, Public Service Co. of Northern Ill.: Power, Jan. 15, 1924, p. 80. Power Plant Engrg. Jan. 15, 1924, p. 119.

Cherry River Paper Co.: Power, Dec. 18, 1923, p. 990.

Marysville Plant, Detroit Edison Co.: Power, May 29, 1923, p. 824.

Hell Gate Station: Power, May 2, 1922, p. 679.

Delaware Station, Phil. Elec. Co.: Power, May 24, 1921, p. 806.

Consumer Co., Milwaukee: Elec. Wld., Dec. 29, 1923, p. 1314.

PROBLEMS

1. If power costs 1.5 cents per kw-hr. approximate the cost of moving 20 tons of coal per hour a horizontal distance of 50 ft., by means of a screw conveyor.

2. Determine the power required to drive a scraper conveyor carrying 50 tons of bituminous coal per hour, sliding blocks to be used. The weight of the chain and flights with sliding blocks is 26 lb. per linear ft., the capacity of the conveyor is 150 tons per hour. The distance between centers of head and last sprockets is 160 ft. and the angle of conveyor with the horizontal is 30 degrees. Speed 50 ft. per min.

3. Determine the power required to drive a pivoted bucket carrier having a capacity of 60 tons of coal per hour; rollers 6 in. in diameter with 1 3/8-in. pins; weight per ft. of empty carrier, 80 lb.; horizontal length of conveyor, 400 ft.; vertical lift, 60 ft.; 4 right angle turns; horizontal length traversed by loaded buckets, 300 ft.; speed of conveyor, 50 ft. per min.

4. Determine the power required to elevate 140 tons of coal per hour by means of a 24-in. belt. Speed of belt, 300 ft. per min ; vertical lift, 30 ft.; length of conveyor between centers, 300 ft. The system contains 3 fixed and 2 movable trippers.

5. A steam-jet conveyor equipped with one 5/8-in. nozzle has a capacity of 8 tons of ash per hr. If the steam pressure at the nozzle is 125-lb. gage, quality of steam 100 per cent, required the lb. of ash removed per lb. of steam when the system is operating at full capacity. Use Napier's rule (equation 280) for calculating weight of steam discharged through nozzle.

6. What will be the cost of conveying 24 tons of ash if the conveyor in Problem 5 is operating at full capacity and other conditions are as follows: Heat value of the coal, 11,500 B.t.u. per lb. as received; overall efficiency of the boiler units, 70 per cent; boiler pressure, 150 lb. gage; feedwater temperature, 180 deg. fahr.; cost of coal, $5 per ton of 2000 lb.; fixed and operating charges other than cost of fuel, 40 cents per 1000 lb. of steam?

CHAPTER VIII

CHIMNEYS[1]

129. General. — A boiler setting is provided with draft for the purpose of properly proportioning air to fuel supply and conveying the products of combustion through the complete setting, including furnace, tubes, economizers, cinder catchers and the like. The term *draft* without qualification in reality signifies *flow*, but in boiler practice it usually refers to the *pressure difference* producing the flow. Draft may be produced mechanically by means of fans, blowers, and steam jets, or thermally by means of chimneys. Stacks or chimneys generally offer the simplest means of conducting the products of combustion to waste; and since the latter must be discharged at a sufficient elevation to prevent their being a public nuisance, the height of stack necessary to effect this result is often sufficient to create the required draft. Even if considerable height must be added to the stack over and above that required to discharge the gases at a given elevation, the extra cost may be considerably less than that incident to mechanical draft operation. For this reason the majority of small and moderately sized steam power plants depend upon chimneys for draft. In large plants equipped with forced-draft stokers and cinder catchers, or where fuel is burned at a high rate of combustion, or where economizers are used for abstracting heat from the flue gases, mechanical draft is commonly employed; but even in these cases a stack is necessary to dispose of the products of combustion.

130. Chimney Draft. — When in operation, a chimney is filled with a column of gases with higher average temperature than that of the surrounding air. As a result, the density of the gases within the stack is less than that of the outer air, and the pressure at the bottom of the column is less inside the stack than it is outside.

The theoretical maximum static draft of a chimney is the difference in weight of the column of heated gas inside the stack and of a column of outside air of the same height. This maximum can be realized only when there is no flow and there is no transfer of heat, or leakage of air into the chimney.

[1] In this text the terms "chimney" and "stack" are used synonymously. Builders occasionally apply the term "chimney" to the masonry and concrete structures, and "stack" to the steel structures.

Let D = maximum theoretical static draft, in. of water.
H = effective height of the chimney, ft.
d_a = mean density of the outside air, lb. per cu. ft.
d_c = mean density of the inside gas, lb. per cu. ft.
0.192 = factor for converting pressure in lb. per sq. ft. to in. of water.

Then

$$D = 0.192\ H\ (d_a - d_c) \tag{66}$$

While equation (66) offers a simple and accurate means of determining the maximum draft pressure for specified densities, it is not easily applied in commercial design because of the number of variable factors influencing the densities. Thus, the density of atmospheric air may be expressed:

$$d_a = \frac{P - hP_v}{0.754\ T_a} + hd_v{}^1 \tag{67}$$

in which

P = observed barometric pressure, in. of mercury at 32 deg. fahr.
h = relative humidity of the air
P_v = pressure of saturated vapor at temperature T_a, in. of mercury
T_a = absolute temperature of the air, deg. fahr.
d_v = density of saturated vapor at temperature T_a and P_v in. pressure, lb. per cu. ft.

Similarly, the density of the chimney gases may be expressed:

$$d_c = Kd_a\ (T_a \div T_c) \tag{68}$$

in which

K = ratio of the density of chimney gas to that of dry air at the same pressure and temperature.
T_c = absolute mean temperature of the chimney gases, deg. fahr.

By combining equations 66, 67, and 68, we may obtain an expression which contains all of the variables except those involving the effect of air currents across the top of the stack, or, in case of absence of wind, the influence of the heated column of gas above the chimney mouth. Some idea of the extreme variation, in general power plant practice, of the influencing factors may be gained from the following summary:

P, the barometric pressure, decreases approximately 1 in. of mercury for every 1000 ft. increase in altitude, and for a given altitude the extreme meteorological variation is as great as 2.0 in. of mercury.

[1] For derivation of this equation see equations (193–5) and (441).

hP_v ranges from 0.03 in. of mercury in extremely cold, dry weather to 1.0 in. on hot, humid days.

T_a, the outside temperature, may range from -10 deg. fahr. or lower to 110 deg. or even higher.

hd_v ranges from 0.00Q032 lb. per cu. ft. in extremely cold weather to 0.0015 in. on hot, humid days.

K ranges from 1.07 for dry fuels high in carbon content, to 0.94 for fuels producing flue gases high in moisture content.

T_c, the mean temperature of the chimney gases, may range from 800 deg. fahr. or more to as low as 200 deg. Temperatures below 350 deg. are seldom experienced except in connection with economizer practice. Because of the increased height of stack necessary to neutralize the reduction in stack temperatures, economizer installations are commonly made with mechanical draft. Attention should be called to the fact that the actual temperature of the chimney gases is not constant but decreases from the flue entrance to the top because of air infiltration and heat losses. This reduction in temperature varies with the type, size, and construction of the stack, temperature difference between the chimney gases and the outside air, velocity and direction of outside air currents, and numerous other factors. Some idea of the drop in temperature for a few specific cases may be gained from inspection of the curves in Figs. 196 and 197.

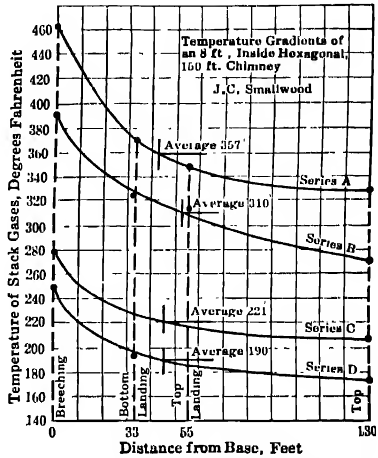
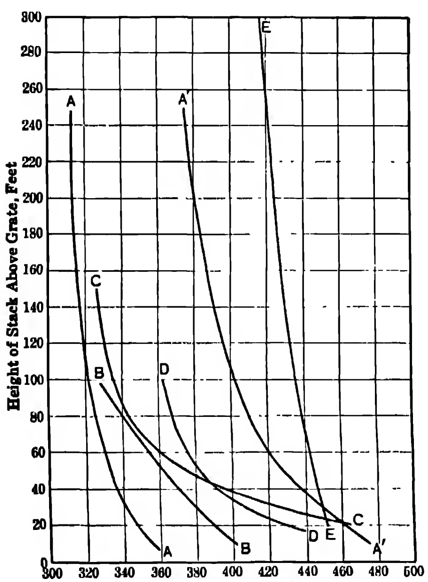


FIG. 196. Temperature Variations in a 150-ft. Chimney at Different Rates of Combustion.



Index	Height, Ft.	Diam., Ft.	Make
AA	250	18	Radial Brick
A'A'	250	18	Radial Brick
BB	100	3	Steel, Unlined
CC	150	8	Radial Brick
DD	102	*3 × 3	Common Brick
EE	332	†7 × 10	Steel, Lined

*Square †Circular Ends

FIG. 197. Temperature Variations in Chimneys.

Loss of Heat in Brick Chimneys: Alfred Cotton, Power Plant Engrg, Aug. 1, 1921, p. 747.

Experiments on Stack Performance: Julian Smallwood, Power, Sept. 16, 1919, p. 464.

Because of the great number of variables and the extreme range in the values of the influencing factors, it is customary, where specific data are not available, to eliminate all but the more important variables: Thus, eliminating hP_v and d_v, and assuming $K = 1$, equations (66) to (68) may be combined and reduced to the following form:

$$D = 0.255\, P\left(\frac{1}{T_a} - \frac{1}{T_c}\right) H \tag{69}$$

An examination of equation (69) will show that for a given set of operating conditions the maximum static draft is independent of the stack diameter and varies directly with the height.

The values in Table 40 are based on equation (69).

TABLE 40

THEORETICAL MAXIMUM DRAFT PRESSURE IN IN. OF WATER, CHIMNEY 100 FT. HIGH

Mean Temp. of Chimney Gases, Deg. Fahr	Temperature of External Air, Deg. Fahr. Barometer 29.92 In. of Mercury										
	0	10	20	30	40	50	60	70	80	90	100
200	0.502	0.467	0.433	0.402	0.370	0.340	0.313	0.285	0.257	0.231	0.206
220	0.536	0.501	0.467	0.435	0.404	0.374	0.347	0.319	0.290	0.265	0.240
240	0.568	0.533	0.500	0.467	0.434	0.406	0.377	0.351	0.323	0.297	0.272
260	0.598	0.564	0.529	0.497	0.466	0.436	0.407	0.379	0.353	0.327	0.302
280	0.623	0.592	0.558	0.525	0.494	0.465	0.436	0.409	0.382	0.356	0.331
300	0.654	0.619	0.585	0.553	0.521	0.492	0.463	0.435	0.409	0.383	0.358
320	0.680	0.644	0.611	0.578	0.547	0.518	0.488	0.461	0.438	0.409	0.384
340	0.704	0.669	0.635	0.603	0.572	0.542	0.513	0.485	0.459	0.433	0.408
360	0.727	0.692	0.658	0.626	0.595	0.567	0.536	0.508	0.482	0.456	0.432
380	0.750	0.715	0.681	0.648	0.617	0.587	0.558	0.531	0.504	0.478	0.454
400	0.771	0.736	0.702	0.670	0.638	0.608	0.579	0.552	0.525	0.500	0.475
420	0.791	0.756	0.722	0.690	0.659	0.629	0.600	0.572	0.545	0.520	0.496
440	0.810	0.775	0.741	0.708	0.680	0.648	0.619	0.592	0.565	0.539	0.514
460	0.828	0.794	0.760	0.728	0.696	0.666	0.637	0.610	0.583	0.557	0.533
480	0.846	0.811	0.778	0.745	0.714	0.684	0.655	0.627	0.601	0.575	0.550
500	0.862	0.828	0.794	0.762	0.733	0.701	0.672	0.644	0.618	0.592	0.567
520	0.880	0.845	0.811	0.778	0.747	0.717	0.688	0.660	0.634	0.608	0.583
540	0.894	0.860	0.826	0.794	0.763	0.732	0.704	0.676	0.650	0.624	0.599
560	0.910	0.874	0.841	0.809	0.778	0.747	0.718	0.691	0.664	0.639	0.614
580	0.925	0.888	0.854	0.823	0.792	0.762	0.734	0.706	0.679	0.654	0.629
600	0.938	0.903	0.870	0.838	0.807	0.776	0.747	0.719	0.693	0.667	0.643
650	0.971	0.936	0.902	0.869	0.838	0.810	0.780	0.752	0.725	0.699	0.675
700	1.002	0.970	0.932	0.899	0.868	0.837	0.809	0.781	0.755	0.729	0.704
750	1.027	0.993	0.959	0.926	0.895	0.865	0.836	0.808	0.782	0.756	0.731

These values are based on the assumption that the chimney gases have the same density as the outside air.

1. For any other height, multiply the tabular quantity by $H/100$ where H is the height in ft.
2. For any other pressure, multiply the tabular quantity by $P/29.92$ where P is the pressure in in. of mercury.

This equation gives results within 5 per cent of those calculated from the more exact laws, for all but extreme conditions — a negligible error considering the probable range in the assumed values for P, T_a and T_c. In applying equation (69) to the design of power plant chimneys, it is common practice to take P as the average barometric pressure for the locality in which the stack is to be built, and T_a as the average temperature of the outside air. T_c may be approximated from curves such as shown in Fig. 197; or, where specific data are not available, it is taken as 0.8 that of the flue gases entering the breeching. See Fig. 65 for influence of rate of driving on flue-gas temperature for a number of types of boilers.

Example 23. — Required the maximum theoretical draft pressure which may be expected from a brick chimney 175 ft. high, by 96 in. diameter, for the following assumed conditions: Barometer, 29.5 in.; temperature of outside air, 60 deg. fahr.; temperature of gases entering base of stack, 550 deg. fahr.

Solution. — Here $P = 29.5$; $T_a = 460 + 60 = 520$; $T_c = 0.8 \times 550 + 460 = 900$ (0.8 = assumed factor for temperature reduction). Substituting these values in equation (69) and reducing

$$D = 0.255 \times 29.5\left(\frac{1}{520} - \frac{1}{900}\right)175$$
$$= 1.07 \text{ in. of water.}$$

As soon as a flow is established, the static draft will decrease, since part of this potential energy is required to impart velocity to the gases and overcome the resistance of the chimney walls. Furthermore, the breeching, boiler, damper, baffles and tubes, and the bed and grate all retard the passage of the gases, and the draft from the chimney is required to overcome these resistances. If an economizer or a cinder catcher is used, this adds a further pressure drop. Neglecting leakage and minor influences, the various pressure losses may be expressed:

$$D = D_g + D_b + D_v + D_d + D_f + D_c + D_r \qquad (70)$$

in which D is the maximum theoretical static draft, D_g the pressure drop through the fuel and grate necessary to effect the desired rate of combustion, D_b the drop through the boiler, D_v the draft pressure required to accelerate the gases from breeching to stack velocity, and D_d, D_f, D_c, D_r, the respective pressure drops through the damper, flue, chimney, and right

angle turns into the breeching. Transposing equation (70) we have

$$D_g + D_b + D_d = D - (D_c + D_f + D_r + D_v) \tag{71}$$

$D_g + D_b + D_d$ is the *static draft required at the stack side of the damper.* $D - D_c$, is the *effective draft of the chimney* and $D - (D_c + D_f + D_r + D_v)$ is the *available draft at the stack side of the damper.*

All of these pressure losses increase approximately with the square of the velocity of flow and may be expressed mathematically; but owing to extreme diversity in operating conditions, many of the factors entering into the analysis can only be roughly approximated, with the ultimate result that the calculated values are more or less arbitrary. Considering the losses in the order given in equation (70):

D, the **total** or **maximum static draft,** may be calculated from equation (69). The limitations of this formula have been previously shown. D_g, the resistance of the fuel bed and grate varies with the kind and condition of the fuel, thickness of fire, type of grate, and efficiency of combustion, and can only be found accurately by experiment. For every kind of fuel and rate of combustion there is a certain draft with which the best results are obtained. The curves in Fig. 60 may be used as a general guide, but the values are only approximate, because the moisture and dust content are not considered. A fuel containing 40 per cent of dust that will pass a 1/8 in. round screen can be burned at only about 60 per cent of the rate which can be secured with the same draft from coal containing only 5 or 10 per cent of dust. Under certain conditions the addition of water to the dust will greatly reduce the fuel-bed resistance. The percentage of solids in coal also affects the draft, as will be seen from inspection of Table 41. D_g does not enter into the chimney design for oil, gas, and powdered fuel, since there is no grate and the fuel is burned in suspension. This is also the case with forced draft equipment in which the fuel-bed resistance is overcome by the fan or equivalent. In certain types of oil, gas, or powdered-fuel furnaces, all or a part of the air for combustion is preheated before it enters the combustion zone. The resistance of the preheating passages to the flow of air may be designated as D_a and should take the place of D_g in general equation (70). Specific data for any type of furnace equipment may be had from the manufacturers.

D_b, the loss of draft through the boiler and setting, varies within wide limits, depending upon the type and size of boiler, arrangement of tubes and baffles, design of setting, type of grate, nature of the fuel, air excess, and rate of driving, and ranges from less than 0.1 in. to 1.0 in. and over. The data given in Table 30 and Fig. 63 may be used as a guide in approximating the extent of pressure drops for different types of boilers and settings. The values in the table apply to hand-fired grates having an air

TABLE 41

INFLUENCE OF SOLID CONSTITUENTS ON THE RESISTANCE THROUGH FUEL BED AND GRATE

(Worker and Peebles)

Natural-draft Stokers

Rate of Combustion Lb. per Sq. Ft. G. S. per Hr.	Pressure Drop Through Fuel Bed and Grate In. of Water — Solid Constituents, Fixed Carbon, Plus Ash, Per Cent				
	40	50	60	70	80
10	0 02	0 03	0 05	0.07	0 10
15	0.07	0 09	0 10	0 13	0.17
20	0 11	0 13	0 16	0 20	0.26
25	0 16	0 21	0 23	0 29	0 36
30	0 22	0 27	0 31	0 38	0 46
35	0.30	0 36	0 41	0.49	0.58
40	0 37	0 43	0.50	0 60	0.70
45	0.45	0 52	0.61	0 72	0.87
50	0 52	0 60	0.71	0.83	1 03

Underfeed Stokers

Rate of Combustion Lb. per Sq. Ft. G. S. per Hr.	Wind-box Pressure, In. of Water — Solid Constituents, Fixed Carbon, Plus Ash, Per Cent				
	40	50	60	70	80
20	0.6	0.7	0.8	1.0	1.3
25	0.8	1 0	1 2	1.4	1.7
30	1.1	1 4	1 6	1.8	2.1
35	1 4	1 7	2 0	2.3	2.5
40	1.7	2 0	2 3	2 7	3.0
45	2 0	2.3	2 7	3.1	3.5
50	2.3	2 7	3.2	3.6	4.1
55	2.6	3 1	3.6	4.1	4.6
60	3 0	3.5	4 1	4.7	5.9
65	3 4	4 0	4.6	5 2	6.0
70	3 9	4 5	5.2	6 0	6.7

space of 45 to 55 per cent and rates of combustion ranging from 20 to 30 lb. of Illinois coal per sq. ft. of grate surface. They also apply to mechanical stokers of the natural-draft type, burning 20 to 40 lb. of coal per sq. ft. of grate surface, with the capacities in either case ranging from rating to 50 per cent overload. The relative pressure drop increases with the load, but there appears to be no close relationship between these two

factors for different boiler equipments. Specific figures may be obtained from boiler manufacturers.

D_v, the draft required to accelerate the gases, varies in accordance with the law

$$h = (V_1^2 - V_2^2) \div 2g \tag{72}$$

in which

h = head in ft. of gas producing the velocity,
V_2 = velocity through the damper opening, ft. per sec.,
V_1 = mean stack velocity, ft. per sec.,
g = acceleration of gravity = 32.2 (approx.).

Assuming a gas density of 0.085 lb. per cu. ft. at 32 deg. fahr. and 14.7 lb. per sq. in. pressure, and reducing head in ft. of gas to pressure in in. of water, equations (69) and (72) give

$$D_v = 0.124 \frac{P_a}{P}\left(\frac{V_1^2 - V_2^2}{T_c}\right) \tag{73}$$

in which

P_a = observed barometric pressure, lb. per sq. in.,
P = one standard atmosphere = 14.7 lb. per sq. in.,
T_c = abs. temperature of the chimney gases, deg. fahr.

The pressure drop necessary to accelerate the gases in their passage through the boiler up to the damper is included in the values assigned for D_g and D_b and hence need not be considered. The draft required to accelerate the gases from the velocity leaving the boiler to that in the stack is ordinarily small and may be neglected, but in case of high velocity differences, 10 ft. per sec. or more, it should be included in the total pressure drop.

D_d, the loss of draft through the damper, is varied arbitrarily to meet the load requirements. The minimum value of D_d corresponding to "wide open damper" is usually included in the boiler loss D_b.

The commonly accepted rules for determining the friction loss D_c through the chimney are all based on Chezy's formula which may be expressed

$$D_c = KV^2H \div DT_c \tag{74}$$

in which

D_c = friction loss in in. of water,
K = coefficient including the coefficient of friction and the various reduction factors,

= 0.008 (This is the algebraic mean of the values assumed by various authorities),

V = velocity of the gases, ft. per sec.,

H = height of stack above the breeching, ft.,

D = diameter of the stack, ft.,

T_c = mean abs. temperature of the chimney gases.

A study of equation (74) will show that, all other conditions remaining unchanged, the draft loss due to chimney friction is in direct proportion to the height. From equation (69) it will be seen that the static draft is also directly proportional to the height. Doubling the height doubles both static draft and the friction loss, but the maximum capacity is unaltered.

When the velocity reaches the point where D_c is equal to the static draft, the maximum capacity of the chimney is attained.

The values in Table 42 are based on equation (74).

Considering *weight* of the gases instead of velocity, equation (74) reduces to the form

$$D_c = k\, W^2 H\ T_c \div d^5 \qquad (75)$$

in which k is a coefficient, including the coefficient of friction and the various reduction constants.

= 1.9 for the constant $K = 0.008$ in equation (74). C. R. Weymouth, Transactions A.S.M.E., Vol. 34, 1912, p. 652, gives k a value of 2.3.

W = weight of chimney gases, lb. per sec.,

d = diameter of the stack, in.

Other notations as in equation (74).

It can be shown[1] that the capacity of the chimney, expressed in weight of gases moved while providing practical draft at the base, is greatest at a little over 600 deg. fahr., and that the capacity falls off when this temperature is exceeded.

D_f, the draft resistance of the flue, or breeching, varies with the size, shape, and construction of the conduit, and may be calculated by means of either equation (74) or (75). In applying these equations to flue calculations, substitute the length of flue for H and the diameter or equivalent for d. The coefficients K and k for the resistance of the flue are ordinarily taken as 20 per cent higher than that for the chimneys. The resistance of square flues is approximately 12 per cent, and that of rectangular flues (ratio 1 1/2 to 1) 15 per cent greater than that of round flues of the same area. A common rule is to allow 0.1 in. of water pressure drop per 100 linear feet of flue. See also equations 123–4.

[1] *Chimney Sizes*, Alfred Cotton, Trans. A.S.M.E., Vol. 45, 1923.

TABLE 42

DRAFT LOSS PER 100 FT. OF A BRICK-LINED CHIMNEY

Barometer 29.92 In. of Mercury
Mean Temperature of Gases 540 Deg. Fahr.

Diameter Ft.	Velocity, Ft. per Sec.								
	10	15	20	25	30	35	40	45	50
4	0.020	0.045	0.080	0.135	0.180	0.245	0.320	0.405	0.500
5	0.016	0.036	0.064	0.108	0.144	0.196	0.246	0.324	0.400
6	0.013	0.030	0.053	0.090	0.120	0.166	0.220	0.270	0.333
7	0.011	0.026	0.046	0.077	0.103	0.140	0.182	0.232	0.286
8	0.010	0.022	0.040	0.067	0.090	0.122	0.160	0.202	0.250
9		0.020	0.036	0.060	0.080	0.109	0.142	0.180	0.220
10		0.018	0.032	0.054	0.072	0.098	0.128	0.162	0.200
11		0.016	0.029	0.049	0.065	0.089	0.116	0.147	0.184
12		0.014	0.027	0.045	0.060	0.082	0.107	0.135	0.167
13		0.014	0.026	0.041	0.058	0.078	0.104	0.131	0.162
14		0.013	0.023	0.038	0.051	0.070	0.091	0.120	0.143
15		0.012	0.021	0.036	0.048	0.065	0.085	0.108	0.133
16		0.011	0.020	0.034	0.045	0.061	0.080	0.101	0.125
17		0.010	0.019	0.032	0.042	0.057	0.075	0.095	0.118
18			0.018	0.030	0.040	0.054	0.071	0.090	0.111
19			0.017	0.028	0.038	0.051	0.067	0.085	0.105
20			0.016	0.027	0.036	0.049	0.064	0.081	0.100

For any height or length H in feet, multiply by 0.01 H.
For any other pressure, multiply by $P/29.92$ where P is in in. of mercury.
For any other temperature t, multiply by 0.001 $(t + 460)$.

D_r, the pressure drop due to right turns, is frequently taken as equivalent to 0.4 the velocity head, and may be calculated from equation (73) by making $V_2 = O$, and substituting 0.05 for the constant. Some engineers assume that the resistance of a turn is equivalent to that of 50 ft. of breeching, and others assume it to be equivalent to the drop in a flue ten diameters long. A rule of thumb is to allow 0.05 in. of water per turn. The discrepancy in results from applying these rules to an assumed set of conditions is decidedly marked. Preference is given to the first rule.

An examination of equations (74) and (75) will show that the friction draft loss of the chimney cannot be calculated directly unless the height and diameter are known. Since these are the quantities to be determined, it is evident that the problem lends itself only to a " cut and try " analysis, provided the equations are to be satisfied. *If the various pressure drops influencing the height of the stack could be calculated or estimated with any degree of accuracy, there would be some reason for exact analysis, but the arbitrary values assigned in practice vary so widely that such analyses are ordinarily without purpose.* Furthermore, the friction loss through the

chimney is only a comparatively small percentage of the total loss (except for high velocities); hence a careful calculation of the chimney friction, along with guesswork in estimating the other losses is highly inconsistent. Scattering tests made on a number of tall chimneys in successful operation show that the **effective** pressure at maximum rating is not far from 80 per cent of the theoretical maximum static pressure. (This factor allows for the drop in temperature of the chimney gases and for the drop in pressure due to friction.) Assuming this to hold true for chimneys in general, the problem of determining the height becomes a comparatively simple one. In view of the uncertainty of many of the influencing factors, results based upon this assumption are perhaps fully as reliable as those calculated from the various formulas, at least for the average plant.

A well-designed central chimney, serving several boilers and subject to considerable load variation, should have comparatively low stack and breeching friction in order to insure "draft regulation." While a certain draft margin is necessary, it should be the aim to provide a chimney with the least possible excess draft over the necessary maximum, future requirements, of course, being considered. For very high stacks, such as are required in tall office buildings, the diameter is made very small so that a considerable portion of the pressure drop will occur in the stack and breeching; otherwise the draft will be excessive even with throttled damper.

The Significance of Drafts in Steam Boiler Practice: U. S. Bureau of Mines, Bul. No. 21, 1911.

Draft and Capacity of Chimneys: Combustion, Mar., 1924, p. 186; May, 1924, p. 354.

131. Chimney Proportions. — A study of equations (69) to (75) will show that any required effective draft may be obtained from various combinations of heights and diameters. Evidently, there must be a certain height and diameter which will produce the cheapest structure. In practice this particular combination cannot be predetermined with any degree of accuracy because of the uncertainty of the various factors entering into the problem of calculating the height and diameters. For an assumed set of conditions, the logical procedure is to calculate a trial area for an arbitrary velocity, and then to proportion the height so that the maximum weight of gases generated may be discharged against the assumed frictional resistances. By "cut and try" a number of combinations of heights and diameters may be calculated which will give the required effective draft. The costs of the various structures may then be estimated and a selection made.

For small stacks, this degree of refinement is seldom attempted, and the usual procedure is to calculate a height compatible with the assumed pressure losses (subject, of course, to community laws), and proportion the area by rules which are more or less empirical.

Example 24. — Proportion a brick-lined stack for water-tube boilers (14 high, vertical three-pass standard baffling) rated at 6000 hp., equipped with natural-draft chain grates and burning Illinois coal; boilers rated at 10 sq. ft. of heating surface per hp.; ratio of heating surface to grate surface, 40 to 1; flue 100 ft. long with two right-angle bends; stack to be able to carry 200 per cent of boiler rating; atmospheric temperature 60 deg. fahr.; sea level; calorific value of the coal 11,200 B.t.u. per lb.; steam pressure 250 lb. gage.

Solution. — A modern plant of this type and size should be able to maintain a combined boiler, furnace, and grate efficiency of approximately 70 per cent at 200 per cent rating.

Maximum b.hp. 6000 × 2 = 12,000.

Heat equivalent of 1 b.hp-hr. = 34.5 × 970.4 = 33,479 B.t.u.

Coal per b.hp-hr. = 33,479 ÷ (11,200 × 0.70) = 4.3 lb. approx.

Total grate surface = (6000 × 10) ÷ 40 = 1500 sq. ft.

Total coal burned per hr. = 4.3 × 12,000 = 51,600 lb.

Maximum rate of combustion = 51,600/1500 = 34.5 lb. per sq. ft. grate surface per hr.

For 70 per cent combined efficiency, the air excess with a natural-draft chain grate and Illinois coal may range from 50 to 75 per cent. To take care of possible reduction in efficiency, leakage, and other adverse influences, assume a total air excess of 100 per cent.

Theoretical air per 10,000 B.t.u. is approximately 7.5 lb. (see Table 12).

Theoretical air per lb. of coal = (7.5 × 11,200) ÷ 10,000 = 8.4 lb.

Actual air per lb. of coal = 8.4 × 2 = 16.8 lb.

Probable weight of flue gas per lb. of coal = 17.5 lb.

If the ultimate analysis of the coal is known, the weight of the products of combustion may be calculated as shown in paragraph (44).

Weight of the flue gas = (17.5 × 51,600) ÷ 3600 = 250 lb. per sec.

Total volume of flue gas = 250/0.044 = 5680 cu. ft. per sec.

The density 0.044 is based on the assumption that the mean temperature of the chimney gases at 200 per cent boiler load is 0.90 of that at the breeching. The temperature of the flue gas leaving the boiler is taken as 518 deg. fahr. See curve *B*. Figure 65.

Assume 25 ft. per sec. as a trial velocity of the chimney gases, then:

Area of stack = 5680 ÷ 25 = 227 sq. ft.

Corresponding diameter, 17 ft.

The various pressure drops at 200 per cent rating may be tabulated as follows:

D_g drop through fuel bed and grate, Fig. 60	0.53 in.
D_b drop through boiler and damper (assumed)	0.80 in.
D_f drop through flue (calculated from equation (74)), assuming resistance of flue as that of an equivalent height of stack	0.03
D_r drop caused by change in direction due to right angle bends (calculated from equation (73)), assuming each turn to have a resistance equivalent to 40 per cent of the velocity head	0.03
D_v acceleration of gases, assuming velocity leaving boiler to be 25 ft. per sec. (equation (73))	0.00
D_t total draft required	1.39

D theoretical maximum draft per 100 ft. of stack, assuming mean temperature of chimney gases to be 0.90 of that of the flue gases leaving the boiler, equation (69)................	0.64
D_c friction drop in stack per 100 ft. of height, equation (74)..........	0.03
D_e effective draft per 100 ft. of stack, $D - D_c$.....................	0.61
H height of stack in ft. above breeching $(D_t \div D_e) \times 100$...........	228

Various combinations of heights and diameters may be calculated in a similar manner by assuming other velocities; thus, for the preceding example:

Velocity, ft. per sec....	20	25	30	35	40
Diameter, ft........	19	17	15.5	14.4	13 5
	Pressure Drops				
D_g fuel bed..	0 53	0 53	0 53	0 53	0.53
D_b boiler	0 80	0 80	0 80	0.80	0.80
D_f breeching....	0 02	0 03	0 05	0 07	0 10
D_r right-angle bends	0 02	0 03	0 05	0 07	0 09
D_v acceleration of gases	0 00	0 00	0 04	0 08	0 13
D_t Total.	1 37	1 39	1 47	1.55	1.65
D draft pressure, 100 ft. of stack	0 64	0.64	0 64	0 64	0.64
D_c friction per 100 ft. of stack...	0 02	0.03	0 05	0 07	0.09
D_e, Effective draft..	0 62	0 61	0 59	0 57	0.55
Height, ft. $(D_t \div D_e) \times 100$	220	228	250	272	300

Any of these stacks will produce the required draft under the assumed conditions, but, other things permitting, the cheapest combination is the one to be selected.

It will be noted that numerous assumptions have been made in the foregoing analysis. Consequently, the reliability of the results depends entirely upon the accuracy of these assumptions. The factors which enter into the problem of chimney draft and capacity are so numerous, their relations so obscure, and values so difficult of numerical determination, that of necessity all chimney equations are more or less empirical; that is, they are simply practice expressed in algebraic form.

For small plants equipped with horizontal return-tubular boilers, the stack proportions in Table 43 are recommended by the Chicago Smoke Department.

For plants of 800 hp. or more, the height of stack for coal burning under natural draft should never be less than 150 ft., regardless of the kind of coal used. Natural draft greater than 1.5 in. of water is seldom necessary,

and higher intensities can be obtained more economically by forced or induced draft. This limits the height of the chimney to about 225 to 250 ft.

TABLE 43

SIZE OF CHIMNEYS FOR RETURN-TUBULAR BOILERS

Boiler Size	Number of Boilers per Stack				No. and Size of Tubes
	One	Two	Three	Four	
48 × 14	$21\frac{1}{2}$ × 90	30 × 100	37 × 110	$42\frac{1}{2}$ × 120	34 – $3\frac{1}{2}''$
54 × 16	$24\frac{1}{2}$ × 95	$34\frac{1}{2}$ × 105	$42\frac{1}{2}$ × 115	49 × 125	34 – 4″
60 × 16	28 × 100	39 × 110	48 × 120	$55\frac{1}{2}$ × 130	46 – 4″
66 × 18	31 × 110	$43\frac{1}{2}$ × 120	$53\frac{1}{2}$ × 130	$61\frac{1}{2}$ × 140	54 – 4″
72 × 18	35 × 120	$49\frac{1}{2}$ × 130	$60\frac{1}{2}$ × 140	70 × 150	70 – 4″
78 × 20	$39\frac{1}{2}$ × 130	56 × 140	68 × 150	$78\frac{1}{2}$ × 160	84 – 4″
84 × 20	45 × 140	61 × 150			

In proportioning the area of the stack on a gas basis, the data in Tables 44 and 45 may be used as a guide. By plotting the data compiled from a number of modern chimneys over 125 ft. in height, the relation between actual velocity at maximum load and diameter appeared to be approximately as follows:

$$V = (0.2 + 0.005D)\ V', \tag{76}$$

in which

V = actual maximum velocity of the chimney gases, ft. per sec.,
D = diameter of the chimney, ft.,
V' = theoretical velocity, ft. per sec.

The theoretical velocity of the gases in the chimney is that developed if the entire static pressure difference is available for producing motion. It may be expressed

$$V' = \sqrt{2\,gh} = 8.03\ \sqrt{H\ (d_a - d_c) \div d_c} \tag{77}$$

in which

h = head of a column of chimney gas, in ft., which would produce the theoretical pressure difference. Other notations as in equations (66) and (75).

For the data in example 24, the theoretical velocity is

$$\begin{aligned} V' &= 8.03\ [175\ (0.07636 - 0.04412) \div 0.04412]^{\frac{1}{2}} \\ &= 90.8 \text{ ft. per sec.} \end{aligned}$$

Chimney Sizes, Alfred Cotton, Mech. Engrg., Sept. 1923, p. 531.
Stack Height to Supply Power, H. Misostow, Power, Oct. 24, 1922, p. 637.
Proportioning of Chimneys on a Gas Basis, A. L. Menzin, Trans. A.S.M.E., Vol. 37, 1915, p. 1065; T. S. Clark, Power, July 29, 1924, p. 175.

132. Empirical Chimney Equations. — Numerous empirical formulas for proportioning chimneys are to be found in engineering handbooks and trade literature. They give satisfactory results within the limits of the assumption upon which they are based, but otherwise may lead to absurd results, their applicability depending largely upon the available data covering the various losses with the particular kind, quality, and condition of coal, and conditions of operation. Occasionally, practical and local considerations fix the height of the stack irrespective of theoretical deductions.

TABLE 44

WEIGHT OF GASES FOR DIFFERENT PERCENTAGES OF CO_2 WHEN CO = 0

(A. L. Menzin)

Per cent CO_2 in the dry gases, by volume	18.7	18.0	17.0	16.0	15.0	14.0	13.0	12.0
Excess air in per cent of the theoretical minimum	0	4.0	10.0	17.0	24.0	33.0	43.0	54.0
Weight of gases per 10,000 B.t.u.	7.8	8.1	8.6	9.1	9.6	10.3	11.0	11.9
Per cent of CO_2 in the dry gases, by volume		11.0	10.0	9.0	8.0	7.0	6.0	5.0
Excess air in per cent of the theoretical minimum		68.0	85.0	105.0	130.0	162.0	206.0	267.0
Weight of gases per 10,000 B.t.u. in the coal		12.9	14.2	15.7	17.6	20.0	23.3	27.8

TABLE 45

AVERAGE VELOCITY OF CHIMNEY GASES

Volume of chimney gases discharged, cu. ft. per sec.	10	100	500	2500	5000	8000	12,000
Average velocity at maximum load, ft. per sec.	10	15	20	25	30	35	40

These values are based upon data compiled from 200 modern chimney installations of various heights and diameters. There appeared to be no definite relationship between volume and velocity, and the values in the table represent gross averages only.

Kent's equation is one of the most popular rules for proportioning stacks for power purposes. It is based on the assumptions that:

1. The velocity of the gases varies as the square root of the height.

2. The retardation of the ascending gases by friction may be considered due to a diminution of the area of the chimney or to the lining of the chimney by a layer of gas which has no velocity, and the thickness of which is assumed to be 2 in.

Thus, for square chimneys,

$$E = D^2 - 8D \div 12 = A - 0.67\sqrt{A} \tag{78}$$

and for round chimneys,

$$E = \pi\,(D^2 - 8D \div 12) \div 4 = A - 0.591\sqrt{A} \tag{79}$$

For simplifying calculations, the coefficient of $\sqrt{A}$ may be taken as 0.6 for both square and round chimneys, and the equation becomes

$$E = A - 0.6\sqrt{A} \tag{80}$$

3. The hp. capacity varies as the effective area E.

4. A chimney should be so proportioned as to be capable of giving sufficient draft to permit the boiler to develop much more than its rated power in case of emergencies, or to permit the combustion of 5 lb. of fuel per rated hp. per hr.

5. Since the power of the chimney varies directly as the effective area E and as the square root of the height H, the equation for hp. for a given size of chimney will take the form

$$\text{Hp.} = CE\sqrt{H}, \tag{81}$$

in which C is a constant, found by William Kent to be 3.33, obtained by plotting the results from numerous examples in practice.

The equation then assumes the form

$$\text{Hp.} = 3.33\,E\sqrt{H}, \tag{82}$$

or

$$\text{Hp.} = 3.33\,(A - 0.6\sqrt{A})\sqrt{H} \tag{83}$$

from which

$$H = (0.3\ \text{Hp.} \div E)^2. \tag{84}$$

The values in Table 46 are based on Kent's equation.

The values in Table 47 are taken from curves plotted by Alfred S. Cotton and give the relative working capacities of chimneys from 5 to 25 ft. in diameter. See Mech. Engrg., Sept. 1923, p. 531. The curves are drawn for a working capacity of 30 per cent of the maximum capacity at 600 deg. fahr. and are based on 90 lb. gas per b.hp. for natural draft, 60 lb. for forced draft and 45 lb. for oil burning.

TABLE 46

SIZE OF CHIMNEYS FOR STEAM BOILERS

Kent's Formula

Diam. In.	Area Sq. Ft.	Height of Chimney, Ft.									
		50	75	100	125	150	175	200	225	250	300
		Commercial Hp. of Boiler*									
18	1.77	23	28								
21	2.41	35	42								
24	3.14	49	60								
27	3.98	65	81								
30	4.91	84	103	119							
33	5.94		130	149							
36	7.07		157	182	201						
39	8.30		190	219	245						
42	9.62		224	258	289	316					
48	12.57			348	389	426					
54	15.90			449	503	551	595				
60	19.64			565	632	692	748				
66	23.76			694	776	849	918	981			
72	28.27			835	934	1023	1105	1181	1253		
78	33.18				1107	1212	1310	1400	1485	1565	
84	38.48				1294	1418	1531	1637	1736	1839	2005
90	44.18				1496	1639	1770	1893	2008	2116	2318
96	50.27				1712	1876	2027	2167	2298	2423	2654
102	56.75				1944	2130	2300	2459	2609	2750	3012
108	63.62				2090	2399	2592	2771	2939	3098	3393
114	70.88					2685	2900	3100	3288	3466	3797
120	78.54					2986	3226	3448	3657	3855	4223
132	95.03					3637	3929	4200	4455	4696	5144
144	113.10					4352	4701	5026	5331	5618	6155

* Based on a consumption of 5 lb of fuel per b. hp. For any other rate, multiply the tabular figure by the ratio of 5 to the maximum expected coal consumption per hp. per hr.

TABLE 47

RELATIVE WORKING CAPACITY OF CHIMNEYS, BOILER HORSEPOWER

(Sea Level and 60 Deg. Fahr.)

(Alfred S. Cotton)

Diameter, Ft.	Coal		Oil	Diameter, Ft.	Coal		Oil
	Natural Draft	Stoker			Natural Draft	Stoker	
5	600	1,000	1,300	15	9,500	14,100	19,000
6	1000	1,500	2,000	16	11,000	17,000	22,500
7	1500	2,100	2,800	17	12,600	19,000	26,000
8	2000	2,900	3,900	18	15,000	22,000	30,000
9	2600	4,000	5,300	19	17,000	25,100	33,500
10	3400	5,100	6,900	20	19,000	28,000	38,500
11	4400	6,500	8,700	21	21,700	32,500	43,000
12	5400	8,100	10,800	22	24,000	37,000	49,000
13	6600	9,800	13,200	23	27,500	41,500	55,000
14	8000	12,000	16,000	24	30,100	46,000	61,000

Working capacity = 28.5 per cent of maximum capacity at 600 deg. fahr. Weight of gases = 90 lb. per hr. for natural draft; 60 lb. per hr. for forced-draft coal, and 45 lb. per hr. for oil, per b.hp.

133. Stacks for Powdered, Liquid, and Gaseous Fuels. — In designing stacks for powdered fuel, oil fuel, or gas firing, the procedure is the same as for coal burning; that is, the height is made sufficiently great to maintain the required draft in the furnace at maximum overload, and the area is proportioned to take care of the maximum volume of gases generated. Excessive draft greatly influences the economy of steam-burner oil-fired furnaces, whereas, with bulk-coal firing, there is rarely danger of too much draft. Consequently, greater care must be exercised in estimating the various draft losses through the boiler and breeching. With oil, gas, and powdered fuel, there is no fuel bed, hence no draft loss on this account, and, because of the smaller air excess required for complete combustion, the pressure loss through the boiler will be less. Furthermore, the action of the burner itself acts to a certain degree as a forced draft. Therefore, both the height and area of the stack for a given capacity of boiler may be less for oil and powdered-coal firing than for bulk-coal firing. Tall chimneys are frequently used in connection with powdered-fuel burning plants, not primarily because of the draft requirements but in order to distribute the flocculent ash at a high elevation. For example, the chimneys at the Cahokia plant are 325 ft. above the burner arches. There are no feedwater economizers in this plant. Table 48, calculated by C. R. Weymouth (Trans. A.S.M.E., Vol. 34, 1912), may be used as a guide in proportioning stacks for oil fuel.

TABLE 48

STACK SIZES FOR OIL FUEL

(C. R. Weymouth)

Stack Diameter, In.	Height in Ft. Above Boiler-room Floor					
	80	90	100	120	140	160
33	161	206	233	270	306	315
36	208	253	295	331	363	387
39	251	303	343	399	488	467
42	295	359	403	474	521	557
48	399	486	551	645	713	760
54	519	634	720	847	933	1000
60	657	800	913	1073	1193	1280
66	813	993	1133	1333	1480	1593
72	980	1206	1373	1620	1807	1940
84	1373	1587	1933	2293	2560	2767
96	1833	2260	2587	3087	3453	3740
108	2367	2920	3347	4000	4483	4867
120	3060	3660	4207	5040	5660	6160

Figures represent nominal rated hp.; sizes as given are good for 50 per cent overloads. Based on centrally located stacks, short direct flues and ordinary operating efficiencies.

With forced-draft stokers, the resistance of the fuel bed does not enter into the calculation for height; otherwise, the procedure in design is the same as for natural-draft coal burning. The values in Table 47 give the relative working capacity of chimneys for bulk-coal (natural-draft and stoker-fired) and for fuel oil.

134. Classification of Chimneys. — Chimneys may be grouped into three classes according to the material of construction:

1. Masonry.
2. Steel.
3. Reinforced Concrete.

The majority of chimneys for power plant service are of common masonry construction, because the materials involved are widely distributed and skilled masons are to be found in almost every community. For very large sizes, special designs of radial brick or tile are preferred to common brick.

Steel chimneys have many advantages and are finding much favor in large power plants, especially where economy of space warrants the erection of the stack over the boiler, in which case the structural work of the boiler setting answers for both boiler and chimney. Among the ad-

vantages over the masonry construction are: (1) ease and rapidity of construction; (2) less weight for a given internal diameter and height; (3) less surface exposed to the wind; (4) lower cost; (5) smaller space required; (6) slightly higher efficiency if properly calked, for there can be no infiltration of cold air as there may be through the cracks in masonry. The chief disadvantage is the cost of keeping the stack well painted to prevent rust, and the corrosive action of the sulphur in the coal.

Reinforced concrete chimneys have many advantages over either the brick or steel constructions, provided they are erected by workmen skilled in the art of concrete mixing and application; but, because of the failure of a few large designs, some engineers are not taking to them kindly.

Steel chimneys may be:

1. Guyed.
2. Self-sustained.

135. Guyed Chimneys. — Guyed sheet-iron or steel chimneys, or stacks held in position by guy wires, are frequently employed on account of their relative cheapness. They seldom exceed 72 in. in diameter and 100 ft. in height. A heavy foundation is unnecessary for the smaller sizes, and the stack may be supported by the boiler breeching. The small, short stacks are ordinarily riveted in the shop, ready for erection, larger sizes being shipped in sections and riveted at the place of installation. In addition to a liberal allowance for corrosion, the material is made heavy enough to support its own weight and to prevent buckling under initial tension of the guy wires and the stress due to wind action. The thickness of shell is ordinarily based on arbitrary rules of practice, and no attempt is made to calculate this value by stress analysis. Table 49 gives the thickness of material as advocated by a number of manufacturers.

TABLE 49

APPROXIMATE DIMENSIONS OF GUYED SHEET-STEEL CHIMNEYS

Height, Ft.	Diameter, In.	Thickness of Shell B.W.G.	Approximate Weight per Ft., Lb.
40	18	16	13
45	20	16	14
45	22	14, 16	20, 15
50	24	14, 16	22, 16
50	26	14	23 5
55	28	14	25
60	30	12, 14	34, 27
65	32	12, 14	36, 28
70	34	10, 12	48, 39
75	36	10, 12	51, 41

Guy wires are furnished in one to three sets of three to six strands each, arranged radially opposite each other, and are attached to angle or tee iron bands at suitable points in the height of the stack. The lower ends of the guys are ordinarily anchored at an angle of 45 deg. with the vertical. A rational analysis of the proper size of guy wires for a specified maximum wind pressure is impracticable because of the number of unknown variables entering into the problem, such as initial tension and stretch of the wires and flexure of the shaft. A common rule is to assume the entire overturning load to be resisted by one strand in each set of guys; thus, if there are two sets of guys the entire load is assumed to fall on two wires. An additional stress of one-half the overturning load is allowed for initial tension. A **lattice bracing** is frequently used between stacks when a number of stacks are placed in a continuous row.

136. Self-sustaining Steel Chimneys. — Steel chimneys over 72 in. in diameter are usually self-supporting. They may be built with or without a brick lining, but the lining is preferred, since it prevents radiation and protects the inside from the corrosive action of the flue gases. Since the independent lining plays no part in the strength of the chimney, it is made only thick enough to support its own weight. In the older designs, the lining is of low-grade fire brick or carefully burned common brick. In these designs the fire brick extends 20 or 30 ft. above the breeching, the remainder of the lining being of common brick. In chimneys up to 80 in. internal diameter, the upper course is 4 1/2 in. thick and increases 4 1/2 in. in thickness for each 30 to 40 ft. to the bottom. In larger chimneys, about 8 in. is the minimum thickness. The lining is generally set in contact with the shell and thoroughly grouted, otherwise depreciation will be very great.

In nearly all recent designs, horizontal rings or shelves of 3 by 4 by 5/16-in. angle iron are riveted to the shell at about 15 ft. centers for supporting the lining. In some designs, vertical stiffeners, which support the horizontal rings, are riveted to the shell. The vertical stiffeners are spaced about 5 ft. apart and the horizontal rings about 20 ft. apart. By either of these methods, any section of the lining may be replaced without disturbing the rest. The lining, usually of vitrified asbestos, is of uniform thickness throughout the length of the shaft and seldom exceeds 4 1/2 in. in thickness.

Self-sustaining stacks are usually cylindrical, though a few designs are tapered; they are generally made with a flared or conical base, the diameter of which is approximately 1 1/2 times the diameter of the stack. The base is bolted to a concrete foundation of sufficient mass to insure stability. In the large, modern station, the stack is frequently carried on a steel structure over the boilers, thereby reducing ground space requirements.

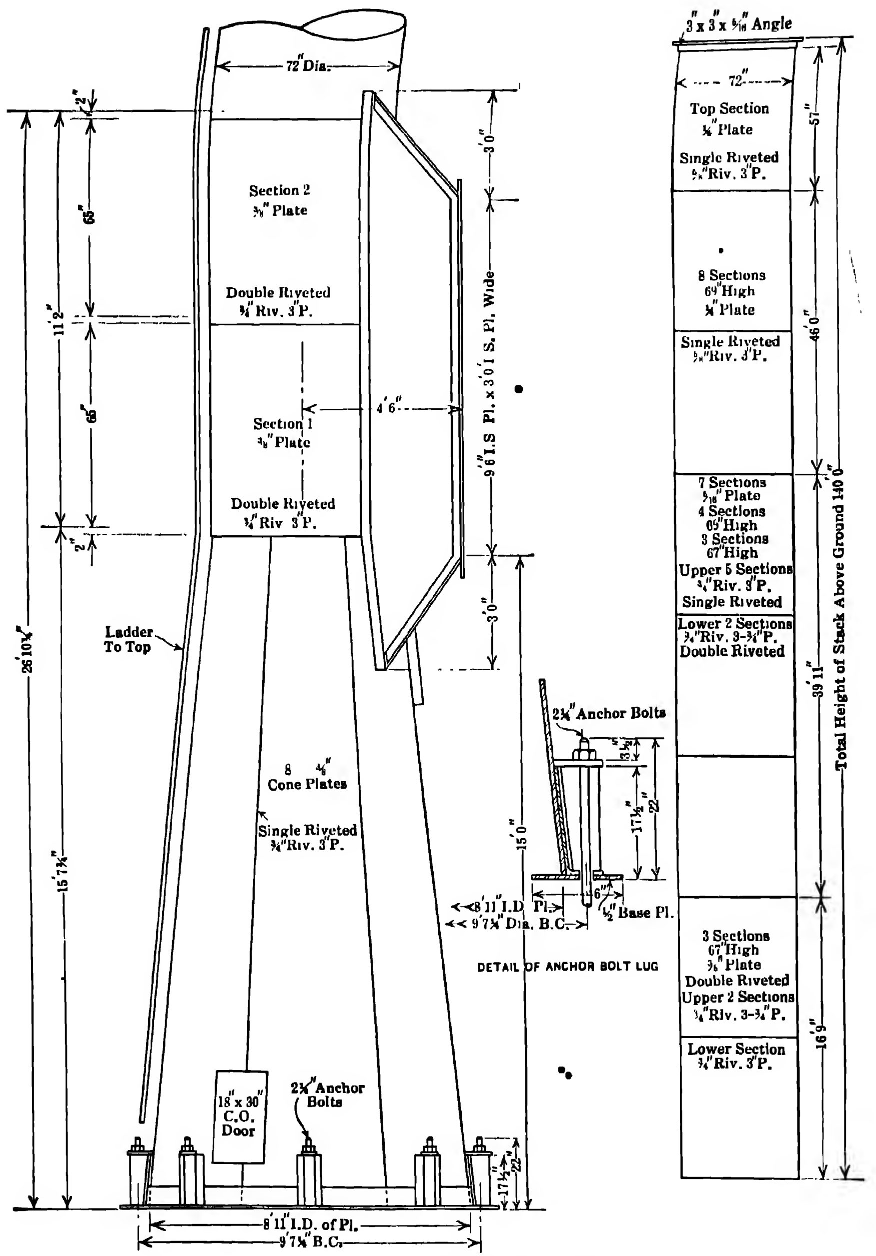


FIG. 198. Details of a Large Self-sustaining Steel Chimney.

Such a design is illustrated in Fig. 175. Every self-sustaining stack should have a ladder, and it is desirable to install a trolley rail for painting purposes. Lightning protection is unnecessary for steel stacks superposed on the structural steel of the building, and ordinarily so for those resting on masonry foundations. In some cases, the base ring for stacks with masonry foundations is connected to a ground plate buried in permanently moist soil.

Figure 198 gives the details of the 140-ft. steel chimney at the power house of the Goldsmith Bros. Smelting & Refining Co., Chicago, Ill., as designed and installed by the Lasker Iron Works, Chicago.

137. Wind Pressure. — Sufficient data are not available to show conclusively the relation between wind velocity and the resulting effective pressure on surfaces of different shapes. Practically all authentic tests have been conducted on small flat surfaces, and there is evidence for the belief that the unit pressure exerted on large surfaces is somewhat less than that obtained from the former. Experiments conducted by different authorities show that the pressure per sq. ft. of flat surface bears the following relationship to the wind pressure:

$$P = KV^2, \tag{85}$$

in which

K = coefficient determined by experiment,
P = wind pressure, lb. per sq. ft.,
V = wind velocity, miles per hr.

The value of K, as determined by the different investigators, varies from 0.0029 to 0.005. The value most commonly used in chimney construction is $K = 0.004$. This corresponds to a pressure of 50 lb. per sq. ft. of flat surface for a wind velocity of 125 miles per hr., the highest allowed for in chimney design. Considering the unit pressure on a flat surface as 1, according to the constants in general use, the effective pressure for the projected area is 0.80 for hexagonal, 0.71 for octagonal, and 0.67 for round columns. Experiments show that the wind pressure increases from the base upward toward the top of the shaft. Christie[1] quotes the following rules as satisfactory for purposes of design:

$$P = P_o + 0.0373\ H \tag{86}$$

$$P_a = P_o + 0.0466\ H' \tag{87}$$

in which

P = average wind pressure throughout the shaft, lb. per sq. ft.,
P_o = pressure at the base of the shaft, lb. per sq. ft.,

[1] *Power*, March 20, 1923, p. 438.

P_a = actual pressure at any given height H', lb. per sq. ft.,
H = height of the shaft, ft.

European designers consider this variation in pressure and allow a value 20.5 to 31 for P_o, but in the United States it is customary to use a single value corresponding to the estimated average velocity throughout the stack. This value ranges from 25 to 30 lb. per sq. ft. of projected area for round stacks. The average pressure allowable is specified by building ordinances in most large cities.

Wind Pressure in Chimney Design: W. Christie, Power, Mar. 20, 1923, p. 438.

138. Thickness of Plates for Self-sustaining Steel Stacks. — If there is no wind blowing, the only stress to be considered in the shell at any section is that due to the weight of the material itself, thus:

$$S_1 = W \div \frac{\pi}{4}(d_1^2 - d_2^2) \tag{88}$$

in which

S_1 = stress (compression) due to the weight of the material, lb. per sq. in. If the shaft is in perfect alignment, this stress is uniformly distributed over the entire cross section under consideration.
W = weight of the shaft above the section under consideration, lb. If the lining is independent of the steel structure, then the weight of the latter only is to be considered; but if the lining is supported by ledges secured to the shaft, then the weight of the lining must be added to that of the steel.
d_1 = external diameter of the tube, in.,
d_2 = internal diameter of the tube, in.

When the wind is blowing, there is an additional stress due to bending. This is a tension on the windward side and a compression on the leeward side, thus,

$$S_2 = Ph \div I/e \tag{89}$$

in which

S_2 = stress in the outer fiber due to wind pressure, lb. per sq. in.,
P = the total wind pressure, lb.,
h = distance from the section under consideration to the center of wind pressure, in. For a cylindrical shaft, h = 1/2 height of shaft above section.
I/e = sectional modulus = $\pi (d_1^4 - d_2^4) \div 32\, d_1$.

The net stress, S, is therefore

$$S = S_1 \pm S_2 = \frac{W}{\frac{\pi}{4}(d_1{}^2 - d_2{}^2)} \pm \frac{Ph}{\frac{\pi}{32}\left(\frac{d_1{}^4 - d_2{}^4}{d_1}\right)}. \tag{90}$$

Equation (90) may be written

$$S = \frac{[W\,(d_1{}^2 + d_2{}^2) \div 8\,d_1] \pm Ph}{\frac{\pi}{32}\left(\frac{d_1{}^4 - d_2{}^4}{d_1}\right)}. \tag{91}$$

$(d_1{}^2 + d_2{}^2) \div 8\,d_1$ is commonly called the radius of the statical moment (see paragraph 146). Designating this quantity by q, equation (91) reduces to the convenient form

$$S = (Wq \pm Ph) \div I/c \tag{92}$$

Because of the liberal factor allowed for the safe working stress, and because a tube of large diameter with thin walls will probably fail by flattening or buckling on the leeward side and not by tension of the windward side, the influence of the weight of the material is ordinarily neglected and the shaft is treated as a cantilever subject to wind pressure only. Wq therefore is neglected, and equation (92) becomes

$$S = Ph \div I/c \tag{93}$$

Since the thickness of the wall is a small fraction of the diameter, the section modulus I/c becomes, approximately,

$$I/c = 0.785\,d_1{}^2t$$

in which

t = thickness of the shell in inches.

Substituting this value in equation (93)

$$S = Ph \div 0.785\,d_1{}^2t \tag{94}$$

A number of steel-stack builders simplify equation (94) still further by making the constant 0.8, thus

$$S = Ph \div 0.8\,d_1{}^2t \tag{95}$$

Considering the stress, S', per linear in., instead of that per sq. in., equation (95) becomes

$$S' = Ph \div 0.8\,d_1{}^2 \tag{96}$$

Example 25. — Determine the thickness of plate at a section 150 ft. from the top of a cylindrical steel stack 12 ft. in diameter and 200 ft. high. Horizontal seams to be single-riveted.

Solution. — The total wind pressure on the section is

$$P = 150 \times 12 \times 25^* = 45{,}000 \text{ lb.}$$

The moment arm is

$$h = 150/2 \times 12 = 900 \text{ in.}$$

S = 8000 lb. per sq. in. (A common allowance for safe stress is 8000 lb. per sq. in. for single-riveted and 10,000 for double-riveted joints.)
Substituting these values in equation (95)

$$8000 = 45{,}000 \times 900 \div 0.8 \times \overline{144}^2 t$$

from which

$$t = 0.305.$$

The nearest commercial size lies between 9/32 and 5/16.

TABLE 50

SELF-SUSTAINING STEEL STACKS

Lasker Iron Works
Chicago

Inside Diameter Ft.	Total Height Ft.	Approximate Weight Lb.	How Made
6	150	43,000	50 ft. of 1/4 in., 50 ft. of 5/16 in., 50 ft. of 3/8 in.
8	150	57,700	do
10	150	71,200	do
8	175	70,000	50 ft. of 1/4 in., 50 ft. of 5/16 in., 50 ft. of 3/8 in., 25 ft. of 7/16 in.
10	175	86,000	do
12	175	105,000	do
10	200	98,700	50 ft. of 1/4 in., 50 ft. of 5/16 in., 50 ft. of 3/8 in., 50 ft. of 7/16 in.
12	200	121,000	do
14	200	137,000	do
12	250	166,800	50 ft. of 1/4 in., 50 ft. of 5/16 in., 50 ft. of 3/8 in., 50 ft. of 7/16 in., 50 ft. of 1/2 in.
14	250	188,000	do
16	250	222,000	do
18	250	242,000	do

Base diameter approximately 1½ diameter of stack.
Cone base arbitrarily set at about 20 ft. 0 in.
All self-supporting stacks to have ladder.
Weights do not include lining or lining-supporting angles.
Eight anchor lugs usually supplied.

* See paragraph 137.

139. Riveting. — The diameter of rivets should always be greater than the thickness of the plate, but never less than 1/2 in. The pitch should be approximately 2 1/2 times the diameter of the rivet, and always less than 16 times the thickness of the plate. Single-riveted joints are ordinarily used on all sections except the base, where the joint should be double-riveted with rivets staggered, although in very large stacks all horizontal seams are double-riveted to give greater stiffness to the shaft.

140. Stability of Steel Stacks. — For stability, the resisting moment $W_t q$ must be greater than the overturning moment Ph_1 (see paragraph 146); that is

$$W_t q > Ph_1,^1 \tag{97}$$

in which

W_t = total weight of the structure, including that of the foundation, and the earth filling over the base, lb.,
q = radius of the statical moment of the foundation base, ft.,
h_1 = distance from the center of wind pressure to the base, ft.,

For a square base, the minimum value of q_1 (see end of paragraph 146) is 0.118 L, but it is common practice to use the maximum

$$q = L/6$$

and the condition for stability is

$$W_t L/6 > Ph_1 \tag{98}$$

Expressed graphically: Lay off GP, Fig. 199, equal to the total wind pressure in direction and amount, and acting at the center of pressure of the shaft; lay off GW, equal to the weight of the stack and foundation; find the resultant GR, and produce it to intersect the base line as at R'; if R' falls within the inner third of the base the stack is stable, provided, of course, that the chimney is properly designed and constructed. Therefore, the heavier the combined weight of the chimney and its foundation, the more stable the structure.

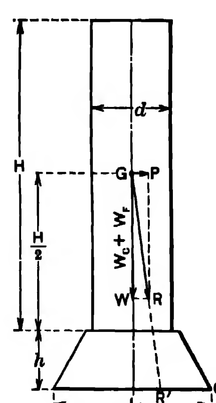


FIG. 199

L in Fig. 199 varies from one-tenth to one-fifteenth H, depending upon the character of the subsoil. (See paragraph 152.)

141. Foundation Bolts for Steel Stacks. — There is no generally accepted rule for proportioning foundation bolts for steel stacks. The

[1] Axis of the shaft assumed to be vertical.

various rules differ principally in the assumed location of the center of moments or neutral axis of the bolts, when stressed by the overturning moment. In the absence of proof to the contrary, and considering the number of unknown factors entering into the problem, the neutral axis may be taken as passing through and tangent to the bolt circle, and the fiber stresses in the bolts may be assumed to be proportional to their distances from the axis. Thus

$$Ph - Wq = SaL, \tag{99}$$

in which

Ph = wind moment at the base ring, in.-lb.,
Wq = statical moment, in.-lb.,
S = maximum fiber stress in the bolts, lb. per sq. in. (To allow for initial stress due to tightening up, a low fiber stress of 12,000 lb. per sq. in. is commonly assumed.)
a = area of each bolt at the root of the thread, sq. in. (All bolts assumed to be of the same diameter.)
L = equivalent mean length of the bolt resisting moment, in.

Referring to Fig. 200, $SaL = S_1b + 2\,S_2c + 2\,S_3d,$ (99a)

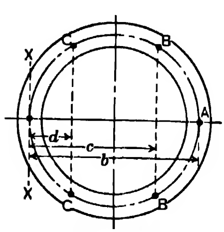


FIG. 200.

in which

S_1, S_2, S_3 = stresses in bolts, A, B-B, and C-C, respectively, lb.,
b, c, d = respective moment arms relative to neutral axis XX, in.

Since the stress in each bolt is assumed to be directly proportional to its distance from the neutral axis, $S_2 = S_1c/b$ and $S_3 = S_1d/b$. Substituting these values in equation (99a) and noting that $S_1 = S_a$, equation (99a) reduces to

$$L = (b^2 + 2c^2 + 2d^2) \div b \tag{100}$$

The value of L becomes

Number of bolts	6	8	10	12	16	24	36
$L = bX$	2.25	3.00	3.88	4.58	6.00	8.90	12.40

Example 26. — Calculate the size of bolts necessary for a steel stack with conditions as follows: Overturning moment 2,750,000 in-lb., bolt circle diameter 82 in., 6 bolts, allowable stress 12,000 lb. per sq. in.

Solution. — Here $Ph - Wq = 2{,}750{,}000$; $S = 12{,}000$; $L = 2.25 \times 82 = 184.5$. Substituting these values in equation (99)

$$2{,}750{,}000 = 12{,}000 \times a \times 184.5;$$
$$a = 1.24 \text{ sq. in.}$$

Nearest commercial size corresponding to this area, $1\frac{1}{4}$ in. diam.

142. Brick Chimneys. — By far the greater number of power plant chimneys are of brick construction and usually of circular section, though octagonal, hexagonal, and square sections are not uncommon. The round chimney requires the least weight for stability, and the others in the order mentioned. They are usually constructed of common or radial brick. Common bricks were used in nearly all the older constructions and are still used in the smaller stacks, but have been almost entirely superseded by the radial product in the modern station.

Brick chimneys are constructed with **single shell,** Fig. 204, and **double shell,** Fig. 202.

The double shell is the more common and consists of an outer shaft of brickwork and an inner core, or lining, extending part way or throughout the entire length of the shaft.

The single shell is the usual construction where carefully burned and selected brick, not easily affected by the heat, are used. As the inner core or lining is independent of the outer shell and has no part in the strength of the chimney, the rules for determining the thickness of the walls are practically the same for both single and double shell. Cast-iron or tile copings are commonly provided on brick stacks to thoroughly protect the top course from the weather.

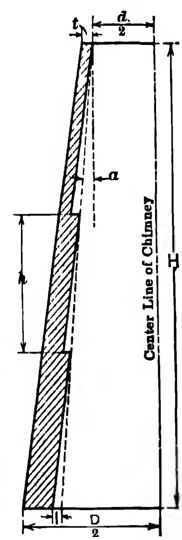


Fig. 201.

143. Thickness of Walls. — The thickness of the wall should be such as to require minimum weight of material for the proper degree of stability, due consideration being paid to the practical requirements of construction. The thickness does not vary uniformly, but decreases from bottom to top by a series of steps or courses as in Fig. 201. In general, the thickness at any section should be such that the resultant stress of wind and weight of shaft will not put the masonry in tension on the windward side or in excessive compression on the leeward side.

For circular chimneys using common red brick for the outer shell, the following approximate method gives results in conformity with average practice:

$$t = 4 + 0.05\, d + 0.0005\, H \qquad (101)$$

where

t = thickness in in. of the upper course, neglecting ornamentation, and should, of course, be made equal to the nearest dimension of the brick in use. Ordinary red bricks measure 8 1/4 by 4 by 2.

d = clear inside diameter at the top, in.

H = height of stack, in.

Beginning at the top with this thickness, add one-half brick, or 4 in., for each 25 or 30 ft. from the top downwards, using a batter of 1 in 30 to 1 in 36.

The minimum value of t for stacks built with inside scaffolding should be 7 in. for radial brick and 8 1/4 in. for common brick, as a thinner wall will not support the scaffold. Radial brick for chimneys are made in several sizes, so that the thickness of the walls, when they are used, increases by about 2 in. at the offsets.

For specially molded radial brick or for circular shells reinforced as in Fig. 202, the length of the different courses may be much less than stated above. The external form of the top is a matter of appearance, and may be designed to suit the taste, but should be protected by a cast-iron or tile cap and provided with lightning rods. Ladders for reaching the top of the chimney are generally located inside the brick stacks and outside the steel structures.

Professor Lang's rule (Engrg. Rec., July 20, 1901, p. 53) for determining the length of the different courses is (Fig. 201):

$$h = C\,(20\,t + 60\,i + 0.1056\,G + 2.5\,d/2 + 656\,tan\,\alpha - 0.007\,H - 0.453\,p - 18.7). \qquad (102)$$

in which

h = length of the course under consideration,

C = constant = 1 for a circular, 0.97 for an octagonal, and 0.83 for a square chimney,

i = increase in thickness for each succeeding section, ft.,

G = weight per cu. ft. of brickwork,

p = wind pressure, lb. per sq. ft.

α = angle of the internal batter.

All other notations as indicated in Fig. 201.

For chimneys over 100 ft. in height, he recommends that 100 be used instead of the actual height, since the critical point will be in one of the lower sections and not at the base.

If a value of h is obtained which is not contained an even number of

times in H, it may be slightly increased or decreased so as to effect this result.

To determine the stresses at any section, the shaft is treated as a cantilever uniformly loaded, with a maximum wind pressure of 25 lb. per sq. ft. If the tension on the windward side subtracted from the compression leaves a positive remainder, the chimney will be under compression throughout the section; if the remainder is negative, the masonry will be in tension, which it withstands but feebly. The sum of the compressive stresses on the leeward side due to wind pressure and weight must be less than the crushing strength of the masonry. The practice, however, of assuming a fixed value for allowable pressure irrespective of the height of the stack gives dimensions that are too low for small stacks and too high for large stacks. According to Professor Lang, compressive stress on the leeward side, in lb. per sq. in., with single chimneys should not exceed

$$p = 71 + 0.65\ L, \tag{103}$$

where

p = pressure in lb. per sq. in.
L = distance in ft. from top of chimney to the section in question.

With double shell,

$$p = 85 + 0.65\ L. \tag{104}$$

The tension on the windward side should not exceed,

$$\text{for single shell;}\quad p = (18.5 + 0.056\ L), \tag{105}$$
$$\text{for double shell;}\quad p = (21.3 + 0.056\ L), \tag{106}$$

Example 27. — Determine the maximum stress in the outer fiber of the brickwork at the base of section 8 of the chimney illustrated in Fig. 204.

Solution. — Assume the weight of the brickwork 120 lb. per cu. ft. and a maximum wind pressure of 25 lb. per sq. ft. of projected surface. The height of the chimney to section 8 is 131.4 ft. The projected area as computed from the figure is 1800 sq. ft. Hence p, the total wind pressure, is $1800 \times 25 = 45{,}000$ lb. The volume of brickwork above section 9 may be calculated, and is 6150 cu. ft., hence the weight $W = 6150 \times 120 = 738{,}000$ lb. The area of the joint at this section is 75.3 sq. ft., therefore, the pressure due to the weight of the superimposed brickwork is 738,000 divided by $75.3 = 9800$ lb. per sq. ft.; $h = 55$ ft. (found by laying out the section and locating the center of gravity); $d_1 = 16.2$, $d = 12.9$. The stress due to the wind pressure may be found by substituting the proper values in equation (89); thus:

$$45{,}000 \times 55 = 0.0983\ S\ (\overline{16.2}^4 - \overline{12.9}^4)/16.2$$

from which $$S = 9907 \text{ lb. per sq. ft.}$$

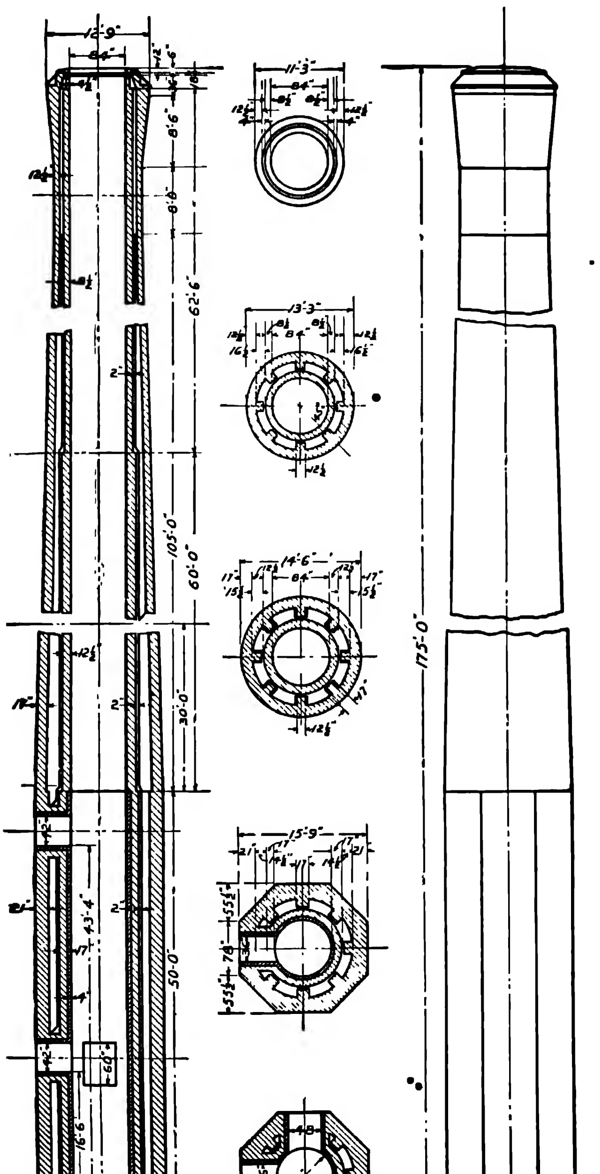


Fig. 202. Brick Chimney at the Power Plant of the Armour Institute of Technology.

The net stress on any part of the section is the resultant of that due to the weight of the stack and that caused by the wind, the net stress on the windward side being

$$-9907 + 9800 = -107 \text{ lb. per sq. ft.}$$

which is evidently a tensile stress and should never exceed the value given by formula (105):

$$p = 18.5 + 0.056\ L = 18.5 + 0.056 \times 131.4$$
$$= 25.8 \text{ lb. per sq. in., or } 3715 \text{ lb. per sq. ft.}$$

The net compressive stress on the leeward side is 9800 + 9907 = 19,707 lb. per sq. ft. which should not exceed that given by formula (103):

$$p = 71 + 0.65\ L = 71 + 0.65 \times 131.4$$
$$= 156.4 \text{ lb. per sq. in., or } 22{,}521 \text{ lb. per sq. ft.}$$

144. Core, Lining, etc. — The core, or lining, of a brick chimney is commonly carried to the top of the shaft, though it sometimes extends only part of the distance. The inside diameter is generally uniform, the offsets being made on the outside. The core and outer shell should be independent, to prevent injury due to expansion of the core. The rules for the thickness of lining in steel chimneys without supporting shelves apply also to brick chimneys. The batters for the inner and outer shells should be such as to allow at least 2 in. clearance between the two shafts at the top, and the top should be protected by an iron ring or by a projecting ledge from the outer shell. **Lightning protection** is always required and usually consists of three or more platinum-tipped copper points connected by a heavy cable to an ample ground plate buried in permanently moist soil. An outside ladder is always a desirable, but not a necessary feature on any type of stack. Modern central station stacks are usually provided with hopper-bottom floors below the flue opening, for collecting and discharging the cinders. In some installations the cinders are removed by steam ejectors, but in most cases the cinders are dumped by gravity and removed by hand trucks or barrows.

145. Materials for Brick Chimneys. — Brick for the external shaft should be hard burned, of high specific gravity, and laid with lime mortar strengthened with cement. Lime mortar itself is more resistant to heat, but hardens slowly and may cause distortion in newly erected stacks, and hence should be used only when a long time is taken in building. Mortar of cement and sand alone is not to be recommended, since it does not resist heat well and is attacked by carbon dioxide, particularly in the presence of moisture. A mortar consisting of 1 part by volume of cement, 2 of lime, and 6 of sand may be used for the upper brickwork; 1, 2 1/2, and 8 respectively for the lower part; and 1, 1, and 4 respectively for the cap.

The harder the brick the more cement is necessary, as lime does not cling so well to hard, smooth surfaces. The inner core may be constructed of second-class fire brick, since the temperature seldom exceeds 600 deg. fahr. Lime mortar is invariably used for the core. In the modern plant, common brick have been almost entirely superseded by radial brick and tile. See paragraph 147.

146. Stability of Brick Chimneys. — When there is no wind blowing and the chimney is built symmetrically about a vertical axis, the pressure due to weight is uniformly distributed over the bearing surfaces, and the center of pressure lies in the line XX, Fig. 203. But when the wind blows, the pressure exerted tends to tilt the shaft as a whole column in the direction of the current, and the resultant pressure at the windward side of the base decreases, until, with a sufficiently high velocity of wind, it may become zero or even negative, in which case the center of pressure moves towards the leeward side of the base. As soon as the pressure at A becomes zero, the joint begins to open (assuming no adhesion between chimney and base) and the shaft is evidently in the condition of least stability. The distance e' from the center of gravity of the shaft to the center of pressure is called the **eccentricity** and may be expressed:

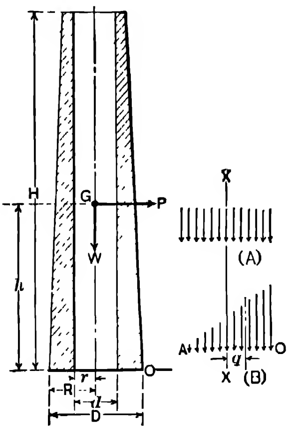


FIG. 203

$$e' = Ph/W \tag{107}$$

in which

e' = eccentricity, in ft.

other notations as in equation (89).

The distance q from the center of gravity of the shaft to the center of pressure for the condition of least stability (i.e., zero stress in the outer fiber on the windward side) is called the radius of the **statical moment** or the **radius of the kern.** The kern itself is the area enclosed by the locus of the center of pressure. Evidently, for stability (assuming that tension is not permissible), the center of pressure must fall within the area of the kern, that is

$$e' \geqq q \tag{108}$$

$$\text{or } Ph \leqq Wq \tag{109}$$

$e' \div q$ is sometimes called the **factor of stability.**

It may be shown that the value of q for the condition of least stability is a function of the cross section only, or

$$q = I/Ae^{*} \tag{110}$$

in which

I = moment of inertia of the section about the gravity axis at right angles to the direction of the wind.
A = area of the section,
e = distance from the center of the shaft to the outer edge of the joint.

Thus, for a solid circular section	q = constant = $D/8$
For a solid square section (maximum)	$q = L/6$
For a solid square section (minimum)	$q = 0.118L$
For a hollow circle	q = constant = $(D^2 + d^2)/8D$
For a hollow square (maximum)	$q = (L^2 + l^2)/6L$
For a hollow square (minimum)	$q = 0.118\ (L^2 + l^2)/L$

A rule of thumb for stability is to make the diameter of the base one-tenth of the height of a round chimney; for any other shape to make the diameter of the inscribed circle of the base one-tenth of the height. Calculations should be made for various sections.

Example 28. — Analyze the chimney illustrated in Fig. 204 for stability at, say, section 8, assuming the weight of brickwork as 120 lb. per cu. ft.

Solution. — From the drawing:
Projected area of the stack, 1800 sq. ft.,
Volume of brickwork, 6150 cu. ft.,
Outside diameter of base, 16.2 ft.,
Inside diameter of base, 12.9 ft.,
Center of pressure to base line, 55 ft.,
Total height above base line, 131.4 ft.
Calculated data:
Maximum total wind pressure:

$$P = 1800 \times 25 = 45{,}000 \text{ lb.}$$

Weight of shaft:

$$W = 6150 \times 120 = 738{,}000 \text{ lb.}$$

Wind moment:

$$Ph = 45{,}000 \times 55 = 2{,}475{,}000 \text{ ft-lb.}$$

Eccentricity:

$$e' = Ph \div W = 2{,}475{,}000 \div 738{,}000 = 3.35 \text{ ft.}$$

* Rankine, "Applied Mechanics" p. 229.

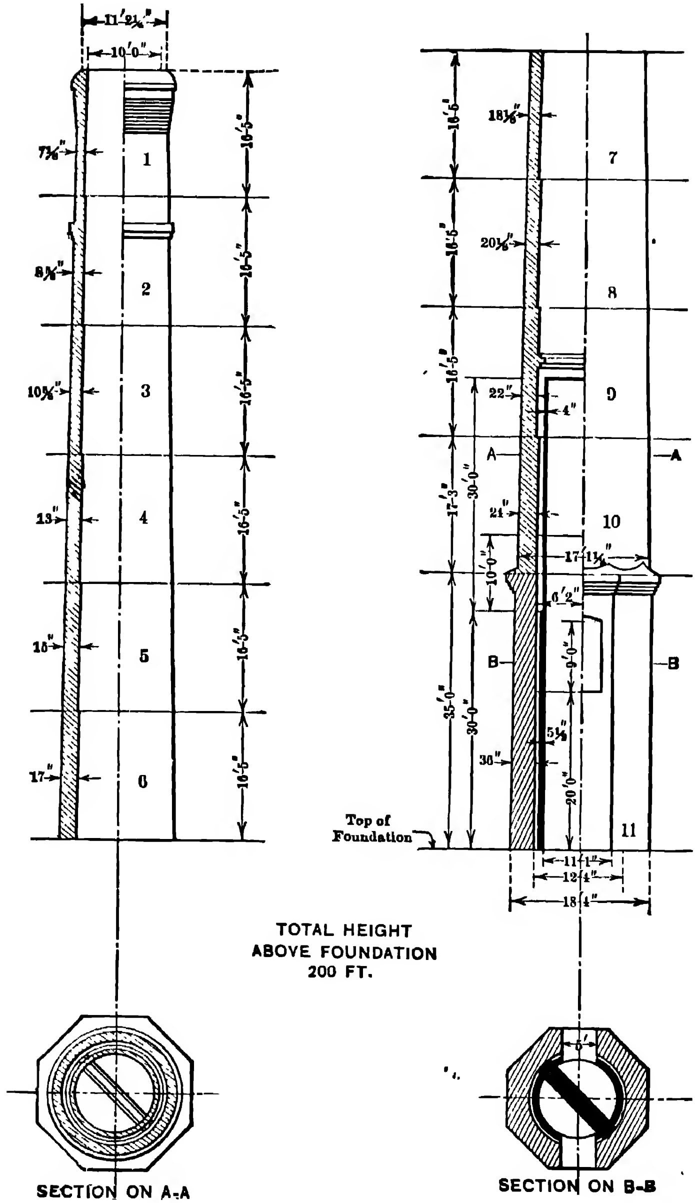


FIG. 204. Custodis Radial Brick Chimney.

Radius of the kern:

$$q = (D^2 + d^2) \div 8D = (\overline{16.2}^2 + \overline{12.9}^2) \div 8 \times 16.2 = 3.3 \text{ ft.}$$

For stability, the radius of the kern should be equal to or greater than the eccentricity. While q is slightly less than e' for this section, the difference is so small that the structure may be considered stable for all practical purposes, particularly in view of the fact that some tension is permissible in stacks of this particular make.

Chimney Design and Construction: T. S. Clark, Power Plant Engrg, Dec. 1, 1920, p. 1121.

The Design of Tall Chimneys: Henry Adams, Ind. Engrg, March, 1912, p. 198.

Self-supporting Chimneys to Withstand Earthquake: C. R. Weymouth, Trans. A.S.M.E., Vol. 42, 1920, p. 787.

147. Radial Brick Chimneys. — Masonry chimneys built of specially molded radial brick are finding increased favor with many engineers because of their many advantages over the common brick structures. The blocks are usually perforated as illustrated in Fig. 205, and are formed to suit the circular and radial lines of each part of the chimney. They are larger than common brick, thereby reducing the number of joints. When the blocks are laid in the wall, the mortar is pressed into the perforations, locking them together in a manner similar to a mortise-and-tenon joint. This, with the breaking of the joints by use of different lengths of radial blocks, forms an excellent bond and greatly increases the strength of the entire structure. An 8-in. wall of radial brick is equivalent in strength to a 12-in. wall of common brick. For ordinary boiler purposes, the lining is approximately one-fifth the height of the stack. The largest chimney in the world at this date (1924) is of this type and is located at Anaconda, Mont. It is 585 ft. high and 60 ft. in internal diameter at the top.

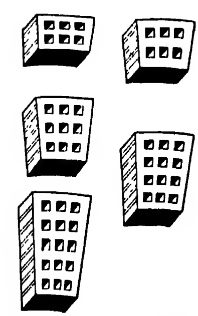


FIG. 205. Custodis Radial Brick.

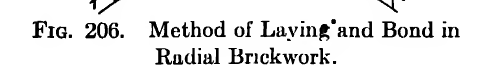


FIG. 206. Method of Laying and Bond in Radial Brickwork.

148. Wiederholt Chimney. — This type of chimney consists essentially of a combination of the masonry and reinforced concrete structures. The inner and outer surfaces of the shaft are formed by vitrified fire-clay tile of special design, as illustrated in Fig. 207. When placed in position, these tile form a permanent mold into which the reinforcing bars and concrete may be introduced. Both vertical and horizontal reinforcing bars

are incorporated in the structure in much the same manner as in the straight concrete type. Because of the tile lining, much higher temperatures may be safely carried than with the concrete type, and the color may be readily made to match that of the power house or adjoining buildings.

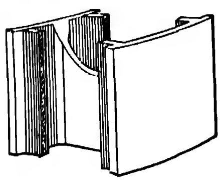


Fig. 207. Tile for Wiederholt Chimney.

149. Reinforced Concrete Chimneys. — Reinforced concrete chimneys have been in use for many years. The advantages claimed for this class of stack are:

1. Light weight, the whole structure being but one-third as heavy as an equivalent common brick chimney. The space occupied is much less than with either brick or steel stack, on account of the thinness of walls at the base and the absence of any flare or bell.

2. Total absence of joints, the entire structure, including foundation, being a monolith.

3. Great resisting power against tension and compression.

4. Rapidity of construction. May be erected at an average rate of 6 ft. per day.

5. Adaptability of the material to any form.

The proper selection of aggregate, scientific mixing of the material, and efficient pouring of the concrete requires greater skill than is frequently employed in fabricating a thin-walled structure such as a chimney; consequently, some of the improperly erected chimneys have been rendered worthless by disintegration and cracking of the concrete. There are many reinforced concrete chimneys in perfect condition after years of service, and this class of structure is finding continued favor with many engineers; but because of the failure of a few of the older designs, some propaganda has been spread concerning their ability to withstand rapid disintegration.

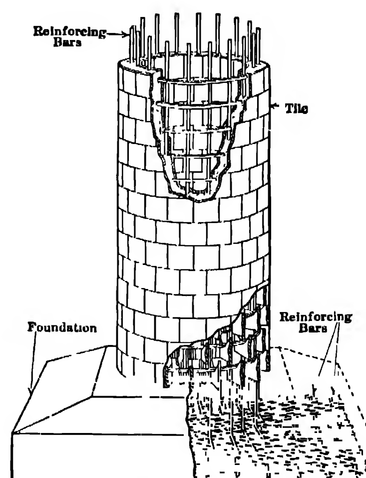


Fig. 208. Method of Reinforcing Wiederholt Tile Chimney.

Figure 209 gives the details of a **Weber "coniform"** steel-concrete chimney as erected at Grafton, Mass., for the Grafton State Hospital. The entire structure, foundation, shaft, and lining, is monolithic, 157 ft. in total height, 7 ft. internal diameter, and weighs only 344 tons. It occupies but 108 sq. ft. of ground space at grade level. The weight of the shaft and lining is 249 tons.

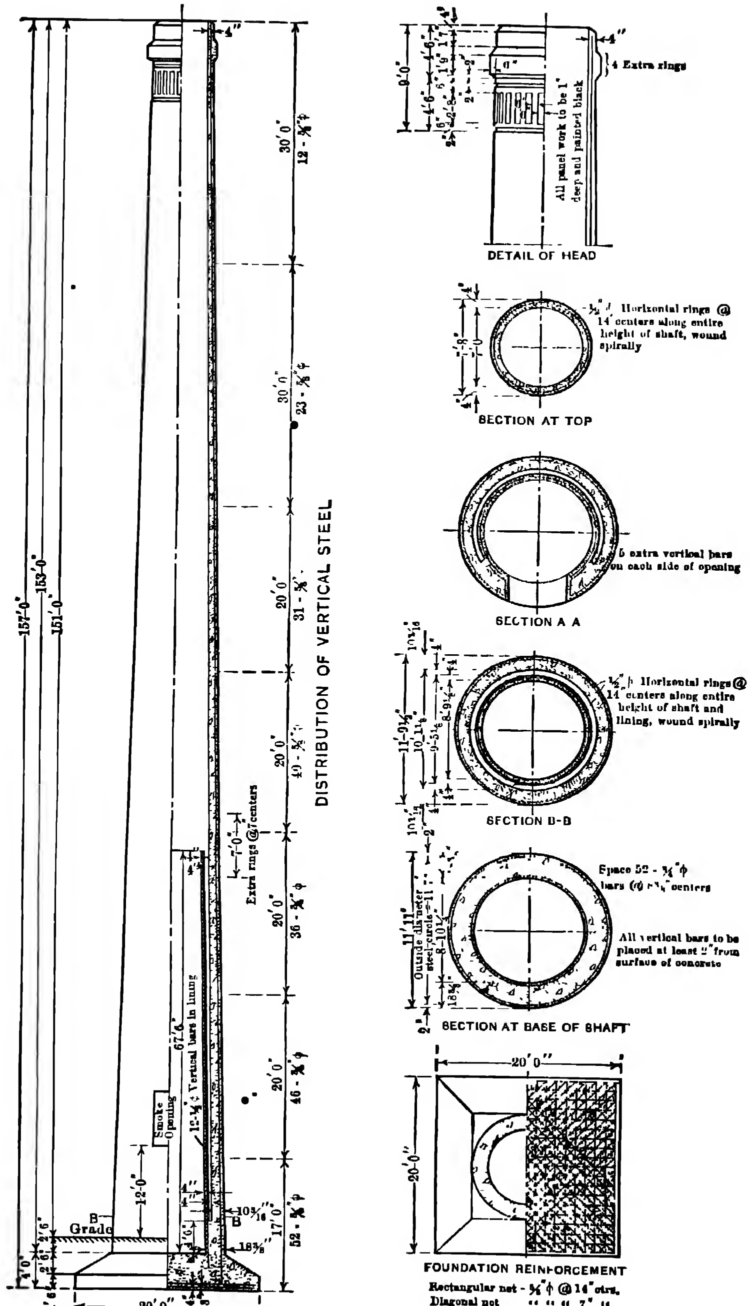


FIG. 209. Weber "Coniform" Reinforced Concrete Chimney.

The shaft is of the double-shell type with inner core extending 65 ft. above the grade. The core is but 4 in. in thickness and the shaft varies from 10 3/16 in. at the junction of the core and shaft to 4 in. at the top. The core reinforcement consists of twelve vertical 1/2-in. twisted steel bars and similar horizontal bars wound spirally at 14-in. centers. The vertical reinforcement in the outer shell varies from fifty-two 3/4-in. twisted bars at the grade to twelve 5/8-in. bars at the top. The horizontal reinforcement consists of 1/2-in. twisted steel rings spaced at 14-in. centers along the entire height of shaft and wound spirally. The steel bars vary from 16 to 30 ft. in length, and where they meet lengthwise are lapped not less than 24 in. The use of different lengths of steel prevents the laps from concentrating in any given section.

One of the tallest chimneys of this type in the world is located in Japan. It is 567 ft. high and 26 ft. 3 in. in diameter at the top.

Lightning protection is almost invariably provided for concrete stacks.

The determination of the amount of steel reinforcement does not permit of simple mathematical calculation because of the number of variables entering into the problem, and graphical charts plotted from semi-rational formulas offer a simple solution. The curves in Fig. 210 are reproduced from "Principles of Reinforced Concrete Construction," 2nd Ed., p. 408, by Turneaure and Maurer, and are used extensively in this connection. The use of the chart is best illustrated by a specific example.

Example 29. — Determine the amount of reinforcement required for the chimney illustrated in Fig. 209 at section BB.

Solution. — From the drawing we find:

D = 11 ft. 9.5 in.	r = radius of the steel circle = 5.79 ft.
d = 10 ft. 1 1/8 in.	h = 153 ft.

The following values may be obtained by simple arithmetic computations, but the actual calculation will be omitted for the sake of brevity.

W, weight of shaft above section BB, 409,000 lb.

A, area of shaft above section BB, 4320 sq. in.

M, wind moment above section BB, 2,600,000 ft-lb.

e, eccentricity $= M/W = 6.36$ ft.

$e/r = 1.1$.

Assume a maximum compression in the concrete of $f_c = 360$ lb. per sq. in. (In practice this assumed value varies from 350 lb. per sq. in. for chimneys under 150 ft. in height to 500 lb. per sq. in. for chimneys 350 ft. high.)

m, a coefficient $= f_c A/W = 3.8$.

From the curves in Fig. 210, the intersection of $m = 3.8$ and $e/r = 1.1$ gives p (per cent of steel required) as 0.53.

But p = area steel ÷ area section.

Whence, area of steel $= 0.0053 \times 4320 = 23$ sq. in., corresponding to 52 3/4-in. steel bars.

Other sections at 20-ft. intervals have been analyzed in a similar manner and the results inserted in Fig. 209.

In the earlier types of steel-concrete chimneys designed and built by

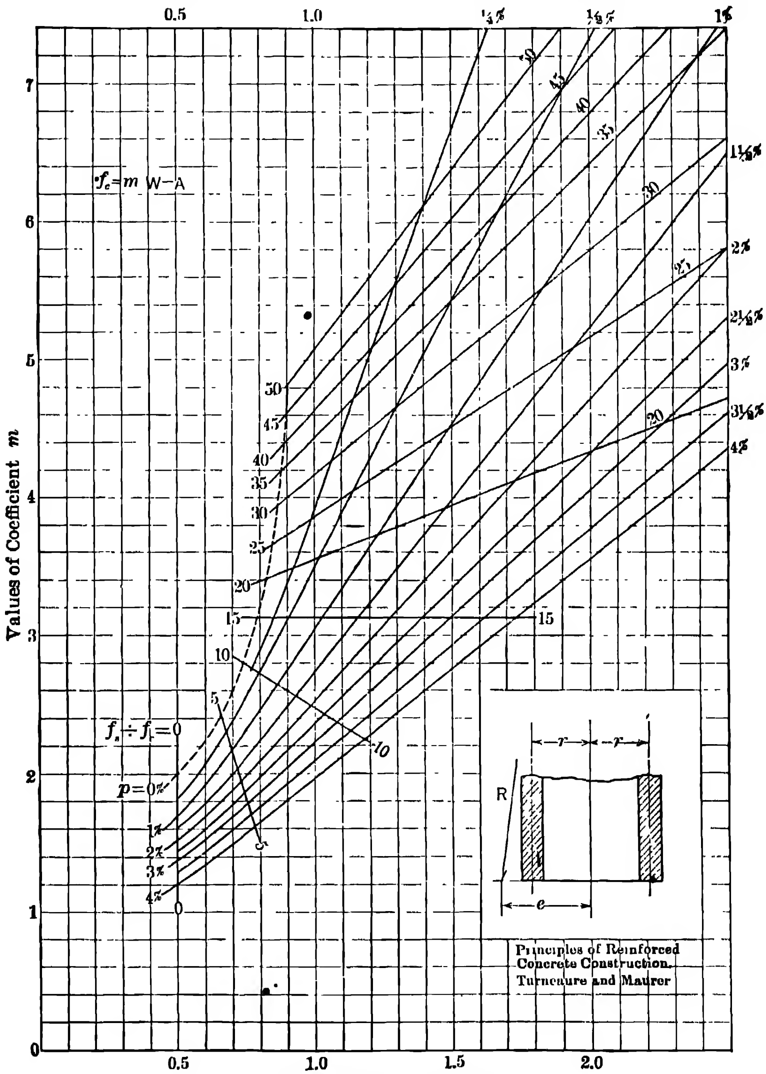


Fig. 210. Wind Stresses in Steel-concrete Chimneys (Turneaure and Maurer).

the Weber Company, the amount of steel reinforcement was calculated from equation (92), but all recent structures are proportioned on the **Turneaure and Maurer chart.** The resultant stress R, as calculated from

equation (92), necessitates the use of more reinforcement than that derived from the chart.

Evasé Stacks. See paragraph 155.

Design, Construction and Cost of a 137-ft. Reinforced Concrete Chimney: Engrg. & Contr., Aug. 11, 1915, p. 111.

150-Ft. Concrete Chimney to Serve Two Breechings: Power Plant Engrg., Feb. 15, 1924, p. 240.

150. Breeching. — The flue or breeching leading from the boiler to the chimney should be proportioned to offer a minimum resistance to the flow of gases, except perhaps where the minimum height of stack is fixed by other than draft requirements. The flue may be carried over the boilers or back of the setting, or even under the fire-room floor. Underground breechings cause excessive pressure drop, and are difficult to clean. For low draft resistance, the breeching should be as short as possible, free from sharp bends, and abrupt changes in area, and of a cross-sectional area approximately 20 per cent greater than that of the chimney proper. While it is possible to find mathematical expressions which permit of rational analysis of the pressure drops due to bends, sudden enlargements, skin-friction and the like, the "coefficients" or experimental factors vary so widely in numerical value, and the probable operating conditions are ordinarily so uncertain, that refined calculations are without purpose. In large installations where the influencing factors may be approximated with reasonable accuracy, the various friction drops are calculated by equations (74) and (75), but, in the majority of small plants, these resistances are based on "rules of thumb," such as allowing 0.10 in. of water pressure per 100 ft. of flue and 0.05 in. per right-angle turn. A breeching of circular cross section causes less draft loss than one of square or rectangular section, and the flatter the rectangle the greater is the draft loss. Clean-out doors should be provided at convenient points for the removal of soot and for access to the breeching. Breechings should be covered with non-conducting material so as to reduce heat losses, and the covering should be on the

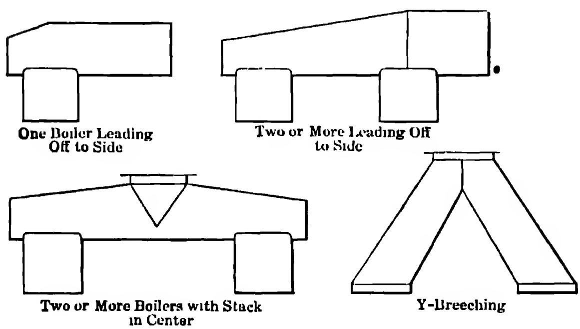


FIG. 211. Types of Breeching Connections for Small Plants.

outside because an inside lining is difficult to repair and deterioration may easily escape detection. The covering material usually consists of 2 in. of block and plastic insulation on wire mesh or rod frame, with a hard cement finish. An expansion joint should be provided in the flue to form a flexible connection between the flue and stack. This is generally located at or near the stack. The flue is also designed to move independently. of the stack, **Gannister** packing being used to effect a seal against air filtration. Each additional boiler connected to the breeching will cause a pressure drop due to friction or interference of the gases as they enter the breeching, or to leakage through the dampers when the boiler is out of service. A common rule is to allow an additional pressure drop of 0.05 for each boiler connected to the breeching. The cross section of the flue need not be the same throughout its entire length, but may be tapered and proportioned to the number of boilers. Where two flues enter the stack on opposite sides, a diaphragm is inserted as indicated in Fig. 204. Some of the different arrangements of breechings and uptakes are illustrated in Fig. 211, and Figs. 212 and 213 give the general details of two recently installed breechings.

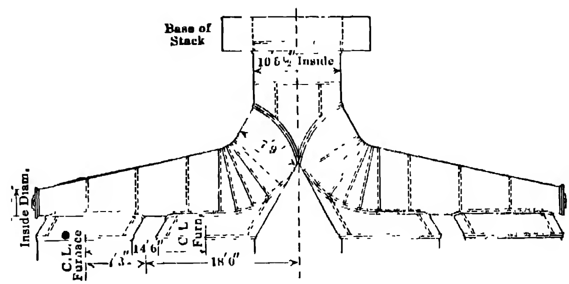


FIG. 212. A Circular Center-connection Breeching for Four Boilers.

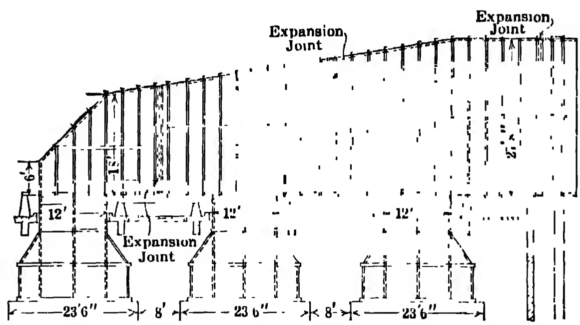


FIG. 213. One Section of a Rectangular Breeching for Six Boilers.

151. Dampers. — Dampers are for the purpose of controlling the rate of combustion by varying the flow of gas to the chimney and for "cutting out" the boilers entirely. Each boiler should be provided with an independent damper for individual control. A main, or stack damper, near the chimney is frequently used for controlling the general or total load. The dampers may be controlled either by hand or automatically (see

paragraph 90). Dampers should be made the full area of the breeching or uptake, and should be preferably installed on heavy horizontal shafts hung on ball or roller bearings at the ends, and with grindstone bearings at intermediate interior points in case of very long shafts. In best practice the end bearings are so installed that they are well ventilated to prevent heating and are protected from cinders and dust.

152. Chimney Foundations. — On account of the concentration of weight on a small area, the foundation of a chimney should be carefully designed. In most cities, the building laws limit the maximum loads allowed for various soils and materials, and although they vary considerably the average range is approximately as follows:

MATERIAL	SAFE LOAD, LB. PER SQ. FT.
Hard-burned brick masonry, cement mortar, 1 to 2	20,000–30,000
Hard-burned brick masonry, cement mortar, 1 to 4	18,000–24,000
Hard-burned brick masonry, lime mortar	10,000–20,000
Concrete, 1 : 2 : 4	20,000–40,000

KIND OF SOIL	SAFE LOAD, TONS PER SQ. FT.
Quicksands and marshy soils	0.5– 1.0
Soft, wet clay	1.0– 2.0
Clay and sand 15 ft. or more in thickness	1.5– 3.0
Pure clay 15 ft. or more in thickness	2.0– 6.0
Pure, dry sand 15 ft. or more in thickness	2.0– 4.0
Firm, dry loam or clay	3.0– 6.0
Gravel, well packed and confined	8.0–10.0
Rock, broken but well compacted	10.0–15.0
Solid bedrock	Up to $\frac{1}{6}$ of its ultimate crushing strength.
	Tons per Pile
Piles in made ground	2.0– 8.00
Piles driven to rock or hardpan	6.0–25.0

Chimney foundations, as a rule, are constructed of concrete, except where the low sustaining nature of the soil necessitates the use of piles or a grillage of timber or steel. For masonry chimneys, the foundation is designed to give the necessary support to the shaft without particular reference to its mass or distribution, as the shape of the foundation has virtually no effect on its stability as a column. In steel and reinforced concrete chimneys, the shape and weight of the foundation are a function of the desired factor of stability, since the shaft is securely anchored to the foundation and the two form practically one mass. The foundation should be designed to fulfill the conditions for shear and flexure in addition to the requirements for stability. Where the foundation is not reinforced, the angle of the sides with the vertical should not exceed 30 deg. The maximum pressure on the soil is the sum of the pressure due to weight

only and that due to the wind moment, or,

$$P_t = 4W_t \div 3b^2(1 - 2e/b) * \tag{111}$$

in which

P_t = maximum pressure due to wind and weight, lb. per sq. in.,
W_t = total weight of the chimney and foundation, lb.,
e = eccentric M/W = wind moment divided by the weight,
b = width of the foundation.

Tall Chimneys: W. Christie, Combustion, Nov. 1913, p. 368.

PROBLEMS.

1. Determine the maximum theoretical static draft obtainable from a chimney 200 ft. high; altitude 2250 ft. (barometer 27.5 in.); temperature outside air 80 deg. fahr.; temperature of the flue gas 500 deg. fahr.

2. What is the maximum theoretical capacity (lb. of gas per hr.) of a chimney 8 ft. in diameter for the following conditions: Mean gas temperature 600 deg. fahr., outside air 60 deg. fahr., sea level, density of gas at 32 deg. fahr. and atmospheric pressure 0.085 lb. per cu. ft?

3. Prove mathematically that the maximum theoretical capacity is independent of height.

4. Calculate the height of an unlined steel stack suitable for burning 20 lb. of Illinois bituminous coal per sq. ft. of grate surface per hr. for a hand-fired return-tubular boiler, standard setting, when the temperature of the outside air is 70 deg. fahr. and that of the flue gas is 450 deg. fahr. Assume a pressure loss in the boiler of 0.02 in.

5. Determine the height and diameter of stack for a battery of Wickes vertical water-tube boilers rated at 4000 hp., equipped with chain grates and burning Illinois screenings; boiler rated at 10 sq. ft. of heating surface per hp.; ratio of heating surface to grate surface, 65 to 1; flue 50 ft. long; stack to be able to carry 100 per cent overload; atmosphere temperature 60 deg. fahr., average barometric pressure 29 in.; temperature of flue gas at overload 650 deg. fahr.; calorific value of the coal 11,000 B.t.u. per lb. Assume pressure drop through boiler from the curves in Fig. 63.

6. Compute the size of stack for the conditions in Problem 5 by means of Kent's equation.

7. Determine the thickness of plates at various sections for a self supporting steel stack of the height and diameter as calculated in Problem 5.

8. Determine the size of foundation for the chimney in Problem 7; firm clay foundation.

9. Design a brick chimney suitable for the data in Problem 5. Analyze the various sections for strength and stability.

* Principles of Reinforced Concrete Construction, Turneaure and Maurer, 2nd Ed., p. 423.

CHAPTER IX

MECHANICAL DRAFT

153. General. — Chimneys are necessary for discharging the products of combustion at an elevation compatible with health requirements or community ordinances. This height is sufficient, in case of the great majority of small power plants, to produce the maximum draft requirements. Small plants, however, which are supplied with fuels requiring intense draft, such as bone coal, low-grade screenings, culm, and the like, seldom have stacks of sufficient height to overcome the resistance of the fuel bed. In many of our large industrial plants, and in practically all our ultra-modern central stations, the resistance to be overcome in forcing the air through the fuel bed, and the products of combustion through the boiler and setting, is far beyond that obtainable with any reasonable height of stack. Furthermore, if a chimney is deficient in draft because of additions to the boiler equipment or increase in load, there is no method of increasing the natural draft except by adding to the height of stack. Artificial, or mechanical draft, has solved the problem for the large station and offers a simple and effective means of furnishing the entire draft requirements or of boosting stack capacity. In a general sense, where the total resistance necessitates draft pressures exceeding 1.5 in. of water, other conditions permitting, it is more economical to use artificial draft. Some idea of the pressure required to effect high rates of combustion in locomotive and stoker-fired stationary plants may be gained from an inspection of the curves in Figs. 214 and 215. See also paragraph 81 for

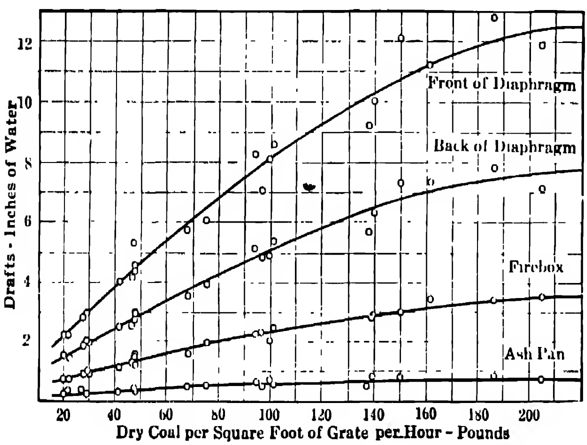


FIG. 214. Relation Between Draft and Rate of Combustion. Consolidated Locomotive.

pressure drops through boilers and paragraph 263 for pressure drops through economizers.

Mechanical draft has many advantages, and under certain conditions is indispensable; it is very flexible and readily adjusted to effect various rates of combustion, irrespective of climatic influences, and permits any practical degree of overload without undue expenditure of energy.

Artificial draft may be broadly classified under two heads:

1. The **vacuum** or **induced draft,** and
2. The **plenum** or **forced draft.**

In the induced-draft system, a partial vacuum is produced above the fire by suitable apparatus, and the effect is substantially that of natural draft.

In the forced-draft system, pressure, above that of the atmosphere, is created below the fuel bed, the air being forced through the fuel.

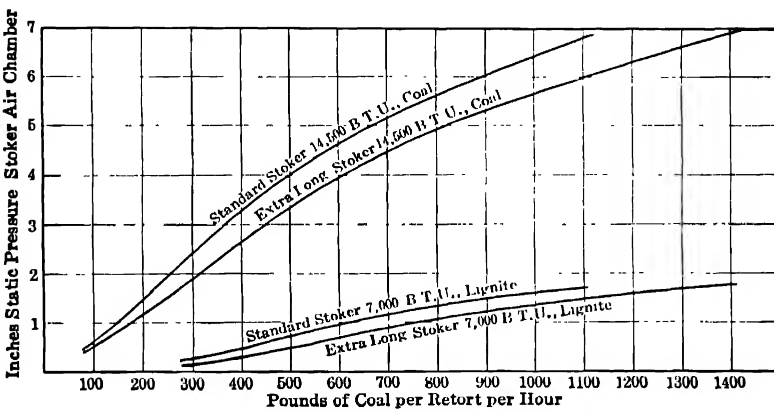


Fig. 215. Approximate Forced-draft Pressure Required at Different Rates of Combustion.

A neutral furnace draft may be effected by a combination of forced draft and induced or chimney draft. The pressure created by forced draft is made sufficient to overcome the resistance of the fuel bed while the chimney or induced draft is depended upon for creating a suction throughout the furnace and setting. The adjustment is such that practically atmospheric or a slight suction pressure exists in the combustion chamber.

In all these systems the artificial draft is usually produced by either (1) **steam jets,** or (2) **centrifugal fans** or **exhausters.**

154. Steam-jet Blowers and Exhausters. — Steam jets are frequently used for forcing air through the fuel bed and occasionally for creating all

or a part of the draft pressure in stationary plants, though this practice is not as common in America as in certain parts of Europe. Steam jets placed in the breeching or stack create a suction throughout the boiler setting, and their action is similar to that of a chimney. When placed beneath the fuel bed, they create a pressure greater than atmospheric, and force the air required for combustion through the fuel. Induced draft produced by steam jets is inexpensive in first cost and in cost of operation, provided the steam used is a waste product, as for example, in locomotive practice, but the cost of operation is usually prohibitive if live steam must be used. In order to develop sufficient chimney action for operating the entire boiler, from 3 to 20 per cent of the live steam generated is required by the jet, depending upon the amount of steam generated, nature of the fuel, character of the equipment, and rate of combustion. Figure 216 shows a simple form of jet device for increasing the draft in a stack, but one which is very extravagant in the use of steam. Higher capacities and lower steam consumptions can be had with a single expanding nozzle surrounded by a series of concentric diffusing cones, as illustrated in Fig. 217.

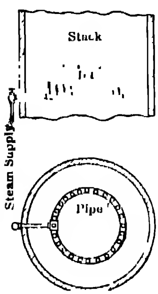


FIG. 216. Ring Steam Jet.

Attention should be called to the fact that it is the *velocity* and not the *weight* of steam which creates the draft, and for this reason the nozzles should be of the expanding type designed for maximum velocity. The suction created by a steam jet for induced-draft service should not exceed 3/4 in. of water; otherwise the steam consumption per cu. ft. of flue gas discharged may become excessive. It is a safe rule to avoid the use of live steam jets for creating induced draft, except possibly in small plants where the stack action is defective and forced draft is inadvisable.

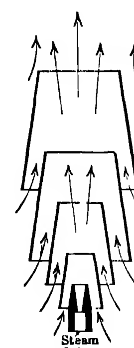


FIG. 217. Principles of Koerting Chimney Ventilator.

Steam jets for forced draft are seldom designed to produce the entire draft requirements of a boiler, but are primarily intended to overcome the resistance of the grate and fuel bed only. In this connection most of them consist essentially of some form of hollow grate bar through which the steam jets force a current of steam and air. Figure 218 shows a type of jet blower which involved to some extent the principle of the ejector. In hand-fired furnaces, the live steam required for jet operation varies from 1 to 16 per cent of the total generated by the boiler, depending upon the amount of steam generated by the boiler; the draft pressure to be devel-

oped; number, size and design of nozzles; design of the grate, and the nature of the fuel and the rate of combustion. The values in Table 51 may be used as a guide for approximating the weight of saturated steam flowing through nozzles of different sizes. For superheated steam see equation (172). In certain types of stoker-fired furnaces, steam consumptions as low as 1.2 per cent of the total generated by the boiler have been recorded. In the latter case the jet merely overcame part of the resistance of the fuel bed. The action of the steam with some classes of fuel is to reduce the formation of clinker and lower the draft resistance of the fuel bed, but a suitable fan in connection with exhaust steam is ordinarily more economical for this condition than a live-steam jet blower. High-pressure live-steam jets are decidedly uneconomical from the standpoint of steam consumption for draft pressures exceeding 3 in. of water, and should never be considered in the design of a new plant, except for purposes other than that of producing "draft." The volume of air delivered by a properly

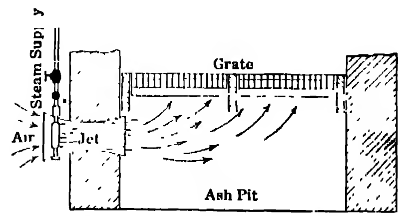


FIG. 218. McClaves Argand Blower.

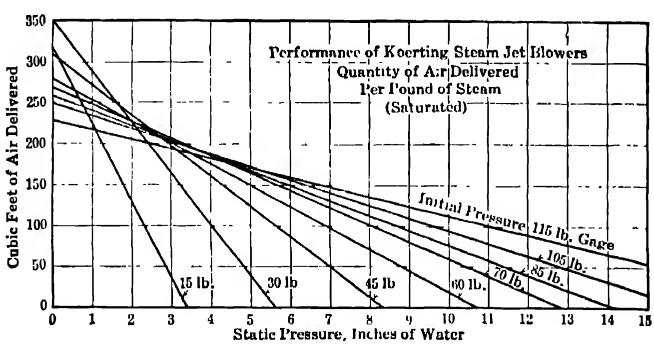


FIG. 219. Performance of Koerting Steam-jet Blowers.

designed jet, with steam at initial pressures varying from 45 to 115 lb. gage, varies from about 250 cu. ft. per lb. of steam for a static pressure of 1 in. of water to approximately 180 cu. ft. per lb. for a static pressure of 4 in. (See Fig. 219.) Water jets serve the same purpose as the steam-jet type of apparatus, but they are seldom found in American practice.

A Modern Steam-jet Furnace: Gas Journal, Sept. 19, 1923, p. 870; Mech. Engrg., Jan. 1924, p. 44.

TABLE 51

APPROXIMATE WEIGHT OF SATURATED STEAM DISCHARGED THROUGH NOZZLES

Lb. per Min.

(Based on Napier's Rule)

Diameter at Smallest Section, In.	Area Sq. In.	Steam Pressure, Lb. per Sq. In. Gage												
		40	50	60	70	80	90	100	110	115	125	150	175	200
1/8	0.0123	0.59	0.70	0 80	0 90	0 99	1 10	1.20	1 31	1.36	1 46	1.73	1.99	2 28
3/16	0.0276	1.33	1 56	1 79	2 02	2 21	2 48	2 71	2 96	3 07	3 31	3 90	4 49	5 09
1/4	0 0491	2 41	2.83	3 26	3 68	3 98	4 39	4 81	5 23	5 44	5 86	6 90	7 96	9 02
5/16	0 0767	3.79	4 45	5 12	5 77	6 23	6 87	7 54	8 20	8 51	9 17	10 75	12 47	14 05
3/8	0.1100	5 42	6 35	7 33	8 25	8 96	9 92	10 85	11 80	12 28	13 23	15 60	17 95	20 35
7/16	0 1503	7.40	8 66	10 00	11 25	12 21	13 49	14 78	16 07	16 72	18 00	21 22	24 43	27 70
1/2	0.1963	9.65	11.30	3 05	14 70	15 92	17 61	19 30	20 98	21 82	23 50	27 70	31 90	36 13
9/16	0.2485	12 25	14 40	16 60	18 70	20 17	22 30	24 42	26 56	27 60	29 75	35 10	40 40	45 75
5/8	0 3068	15 10	17 75	20 40	23 00	24 90	27 55	30 16	32 80	34 10	36 70	43 31	49 89	56 50
11/16	0.3712	18 25	21 40	24 65	27 80	30 15	33 31	36 45	39.68	11 25	44 40	52 40	60 36	68 31
3/4	0.4418	21 80	25 60	29 40	33 20	35 85	39 60	43 43	47 20	49 20	52 90	62 30	71 83	81 30
13/16	0 5185	25 50	29.90	34 40	38 20	42.10	46 50	50 90	55 40	57 64	62 00	73 10	84 25	95 45
7/8	0 6013	29 60	34 80	40 00	45 10	48 75	53 96	59 10	64 27	66 84	72 00	81 80	97 77	110 50
15/16	0 6903	34 00	39 90	46 00	51 75	56 10	61 90	67.80	73 70	76 74	82 60	97 50	112 24	127 10
1	0 7854	38.60	45.40	52 20	59 00	63 75	70 40	77 22	83 94	87 31	93 90	110 82	127 71	144.50

155. Fan Draft. — In the great majority of steam plants operating with mechanical draft, forced or induced, the draft pressure is created by some sort of centrifugal blower or exhauster. A few years ago it was common practice to install a single blower or exhauster for an entire battery of boilers, duplicate fans being installed only where continuity of operation was of prime consideration. While this practice is by no means obsolete, most of the modern boiler units are equipped with independent blowers and exhausters. This is true not only for the huge stoker-fired units in the large central station and industrial plant, but also for the hand-fired boilers in the small isolated station. A common duct or plenum chamber is frequently used in connection with the individual fan system, but this is intended primarily as a "cross over" for emergency use rather than a distributing main.

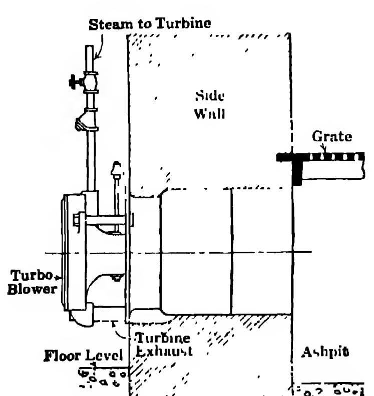


FIG. 220. Typical Forced-draft Equipment. Hand-fired Boiler.

Figure 220 shows an installation of a turbo-undergrate blower in the

side wall of a hand-fired boiler, illustrating current forced-draft practice for this class of equipment. The blower, consisting essentially of a small impulse steam turbine, direct connected to a specially designed propeller fan, may be placed in the rear or front wall instead of the side wall as illustrated. The blower discharges below the grate and may be automatically controlled by damper regulation. The turbine exhaust may be discharged into the ashpit or it may be used in the feedwater heater or other heating devices. While the ordinary propeller type of undergrate blower is a comparatively low-efficiency machine, small blowers of the Coppus "Vano" type have been developed to a high state of efficiency comparable with that of the largest multi-vane units.

Figure 221 shows the application of a forced-draft fan to a boiler unit equipped with underfeed stokers, illustrating current practice.

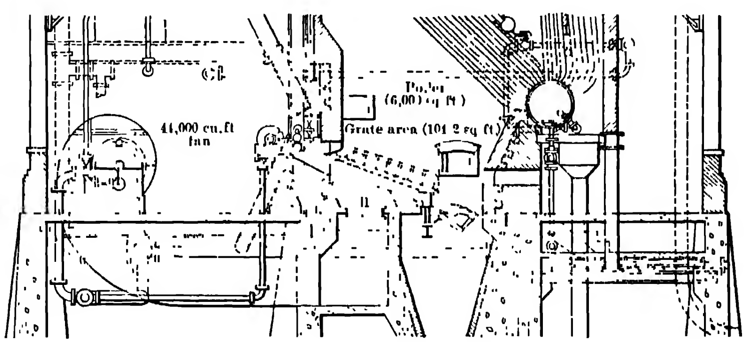


FIG. 221. Typical Forced-draft Installation. Underfeed Stokers.

In most installations, forced draft is employed merely to overcome the resistance of the fuel bed and not to overcome the additional resistances of the gas passages. In order to force the gases through the various boiler passes, as well as to feed air through the fuel bed, a pressure greater than atmospheric would have to be maintained throughout the entire setting. This would tend to force the gases outward through any leaks in the setting, or in case of a tight setting, to force them through the fire door when the fire is cleansed or replenished. In the modern plant a neutral or preferably a slight suction is maintained in the furnace, and the rest of the gas passages leading to the stack are under suction. Air pressures necessary to overcome the resistance of the fuel bed and stoker vary from a fraction to 8 in. depending primarily upon the nature of the fuel, design of stoker, and rate of driving.

Forced-draft traveling-grate stokers are generally furnished with air from a unit-fan system with ducts leading to the various pressure com-

partments, though a central fan system with a main duct leading to the individual boilers is not uncommon, particularly in the smaller plants. The Illinois forced-draft traveling-grate stoker is frequently equipped with a number of small Coppus Turbo-vane blowers, one blower for each compartment.

Figure 222 shows the application of a forced-draft fan to the air-preheater equipment at the Colfax Station of the Duquesne Light Co., Pittsburgh, Pa. The connection between the common forced-draft duct and windbox normally supplying the boilers has been retained, but it is cut off by means of a shut-off damper when the preheater system is in operation. The air is taken into the preheater from the boiler room directly over the boiler, as indicated, carried down the duct by the fan, and discharged into the stoker wind box by means of two ducts extending on either side of the boiler.

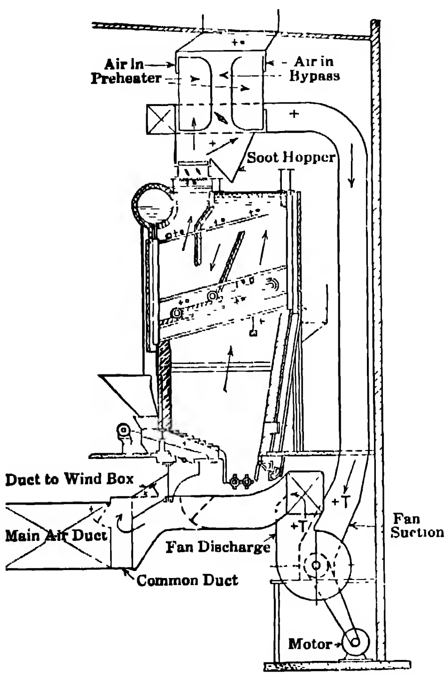


FIG. 222. Forced-draft Stoker Installation with Air-preheater. — Colfax Station.

In marine practice, forced draft is commonly furnished on either the **closed stokehold** or the **Howden** system. In the former the boiler room is entirely closed and provided with air locks for the passage of the boiler-room crew. The fans discharge directly into the boiler room and maintain a static pressure of from 3/4 to 3 in. of water. In the Howden system, the air in the stokehold is at the same pressure as the outside air, but the ashpits are sealed. Most of the air, preheated by the flue gases, is delivered to the ashpits under pressure of 1 to 3 in. of water, but a small amount is admitted above the fires.

In the **induced-draft** system, the suction side of the fan is connected with the uptake or breeching of the boiler or batteries of boilers, and the products of combustion pass through the body of the fan. The action of the induced system is identical with that of a stack of equivalent capacity.

Induced draft is generally necessary in connection with economizers or flue-gas air-preheaters, because the high frictional resistance of these ap-

pliances and the low temperature of the waste gases effected by the heat transmission require an excessive height of stack. Induced-draft fans are also installed in connection with forced-draft blowers where there are

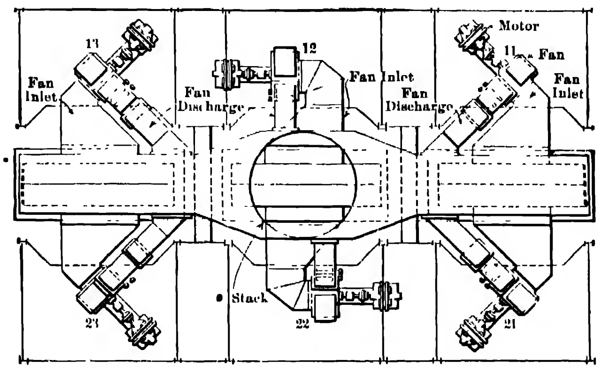


FIG. 223. Induced-draft Equipment for One Group of Boilers. Hell Gate Station.

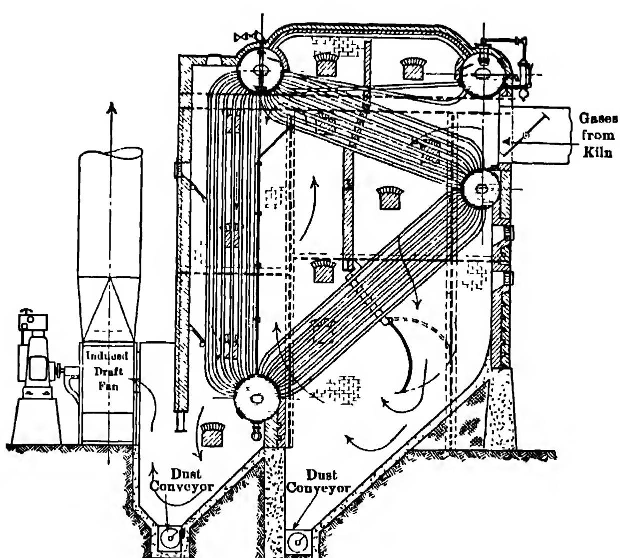


FIG. 224. Induced Draft for Waste-heat Boiler Equipment.

no economizers or air-preheaters, but where the pressure drop from furnace to uptake is very high at peak loads. For example, in the Hell Gate power house of the United Electric Light and Power Company, induced draft is used in connection with a stack 258 ft. high, and there are no

economizers. Each group of six boilers has a single stack and breeching, Fig. 223. Cinder-catching compartments are installed in each uptake. In each of these, the gases rising from the uptake are sharply deflected by a baffle, so that the cinders are thrown into a tank of water. The induced-draft motors are started by hand but are thereafter automatically regulated by balanced-draft control. Provision is made to by-pass the flue gases around the induced-draft fans inasmuch as these fans are needed only at peak loads.

Figure 224 shows the application of an induced-draft fan to a Kidwell boiler setting for utilizing waste heat from cement kilns.

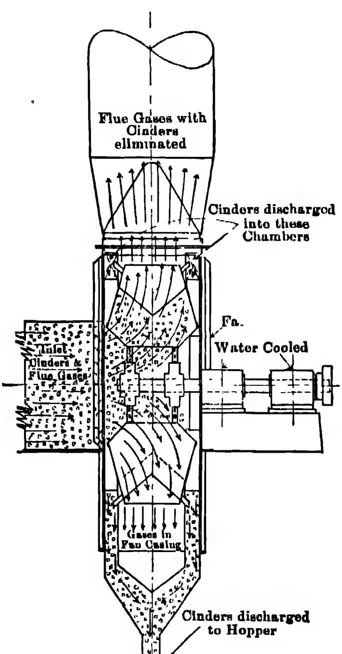


Fig. 225. Sturtevant Combined Induced-draft Fan and Cinder Eliminator.

Induced-draft fans operate with much higher gas velocities than forced-draft fans, at times reaching 60 ft. per sec. At these high velocities, cinders and other suspended earthy matter produce a decided erosive action. Specially designed fans for ejecting the cinders are found in the latest power house designs. See Fig. 225. Bearings are also water-cooled to prevent the lubrication from being burned out by the heat conducted from the gases to the journals.

It is not generally appreciated that more power is required to create the draft from the hot than from the cold end, although the weight of gas and the pressure head is the same. An inspection of equation (117) will show that the horsepower is a direct function of the volume of gases discharged, and, since the volume of the flue gas is approximately twice that of the entering air in the average plant, the power required by the induced-draft fan will be about twice that absorbed by the forced-draft fan. The temperature increase of the flue gas by the fan compression will be nearly twice that of the cold air for the same pressure range, thus accounting for the apparent anomaly between horsepower as calculated on the weight and on the volume basis.

The **Ellis and Eaves** system is an application of induced draft in which the air for combustion is preheated by the waste gases before passing through the furnace. The boiler and setting is entirely closed, so that the

suction of the induced draft draws the cold air through the preheater and the products of combustion, in series, through the combustion chamber, boiler tubes, and uptake before discharging them to waste. This system is used chiefly in marine practice.

In the **Pratt** system, induced draft is effected in connection with an **evasé** or **Venturi** stack (1) by introducing a blast of atmospheric air from a low-pressure blower just below the throat of the stack, or (2) by passing a small amount of the chimney gas through an induced-draft fan and discharging it into the throat. The evasé stack is of light sheet-iron construction, comparatively short, and shaped as shown in Figs. 226 and 227. The suction draft is created by the ejector action of the jet in being discharged through the narrow section of throat of the stack. This system is finding favor with European engineers, but has not yet been introduced to any extent in the United States. The arrangement shown in Fig. 227 necessitates the use of higher air pressures and requires more power for a given suction pressure and capacity than does that shown in Fig. 226. In the cold-air system, the static pressure of the blower is approximately eight times the draft-pressure requirements in the breeching. A notable installation of the Pratt system is at the Fulton St. Heating Plant, Grand Rapids, Mich. For a description of this plant, together with guaranteed performance of the stacks and fans, consult *Power*, July 8, 1924, p. 46. The data in Table 52 are of interest in showing the performance of one of the two evasé stacks, 68 ft. high by 9 ft. 10-in. diameter (at the top), at the power plant of the City of Edmonton when operating at various loads. These stacks are 5 ft. 3 in. in diameter at the throat and furnish the draft for eight 4780 sq. ft. B. & W. boilers equipped with chain grates of 100 sq. ft. grate area each.

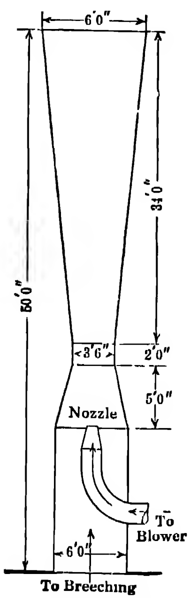


FIG. 226. Evasé Stack. Outer Circuit (Capacity 6000 lb. coal per hr.).

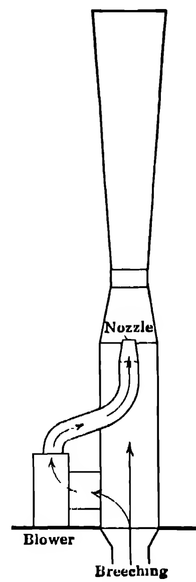


FIG. 227. Evasé Stack. Inner Circuit.

Venturi Stacks, by A. W. H. Griepe; Combustion, Apr. 1922, pp. 166–175.

TABLE 52

PRATT INDUCED-DRAFT PLANT

Fan R.p.m.	Flue-gas Temp. Deg. Fahr.	Suction Pressure In. of Water	Static Pressure In. of Water	Hp. to Operate Fan
456	450	0 71	3 95	26.2
490	450	0 76	4 65	31 6
545	450	0.84	5 50	40 0
600	450	0.98	7 10	49.6
696	450	1 09	8.60	71 0

Tall chimneys are a necessity in most cities, since legislation requires the gases to be discharged at a height above that of adjacent buildings. In such situations, with stokers of the forced-draft type, tall stacks or induced draft would at first thought appear to be a necessary evil. Experience, however, shows that suction draft is an important factor in effecting efficient combustion and in prolonging the life of the furnace brickwork. By mutually adjusting the pressure created by the forced-draft apparatus and the suction of the chimney or its equivalent, a **neutral** or **balanced effect** can be produced in the combustion chamber; that is, the pressure in the combustion chamber becomes practically atmospheric. The relative pressure drops are shown graphically in Fig. 228. This condition of positive pressure under the fire bed, zero or preferably a slight suction pressure in the combustion chamber, and a suction pressure throughout the rest of the setting (1) prevents discharge of the furnace gases into boiler room through leaky fire doors, inspection doors, and cracked setting; (2) minimizes stratification and short-circuiting of the air supply and combustible gases; (3) reduces the "soaking up" action of heat by the furnace brickwork; (4) assists reduction of air excess; and (5) effects an increase in overall boiler, furnace, and grate efficiency. Most

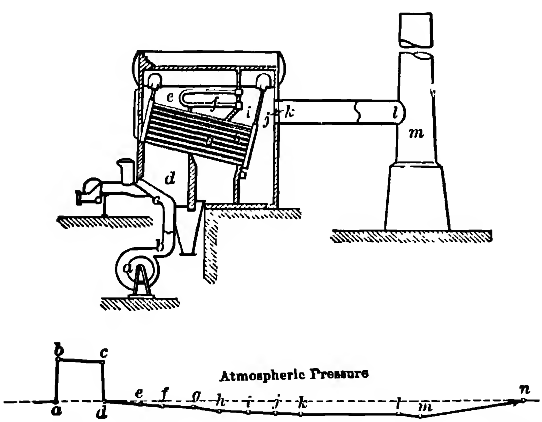


FIG. 228. Pressure Drop through Boilers. — Combined Forced-draft and Chimney.

of our modern central stations are operating with practically balanced-draft conditions. In these plants the stoker speed, fan speed, and stack damper are automatically controlled so as to effect the desired result.

In a number of ultra-modern central stations, the chimneys are 250 ft. high or over, and are served with both forced- and induced-draft fans. The induced-draft fan gives a maximum suction in the uptake of 2 in. or more of water pressure, and the forced-draft equipment is capable of maintaining a static pressure of 10 in. of water under the grates. After the gases have passed from the boiler, this may be discharged directly into the stack, or, by closing proper dampers in the breeching, can be made to pass through the economizer and then to the stack; by closing a second damper, the gases are made to pass through the induced-draft fans before going to the stack. This makes it possible to operate the boilers under the most economical conditions at all times.

Forced Draft (Serial): Steam Power, May to Nov , 1923.
Mechanical Draft: Power Plant Engrg., Oct. 1, 1922, p. 939.

156. Types of Fans. — The large majority of centrifugal fans for mechanical draft may be divided in two general classes: those having rotors with a few straight or slightly curved blades of considerable length radially, Fig. 229, commonly designated as **steel-plate** or **paddle-wheel** fans, and those having rotors with a number of short curved blades, Figs. 230–232,

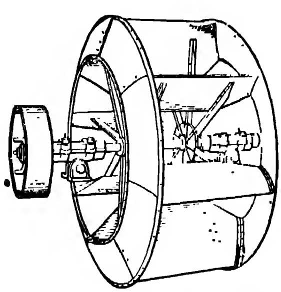


FIG. 229. Standard Steel-plate Fan Wheel.

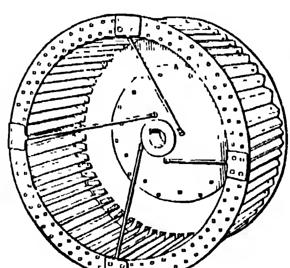


FIG. 230. "Sirocco" Wheel Turbine Type Impeller.

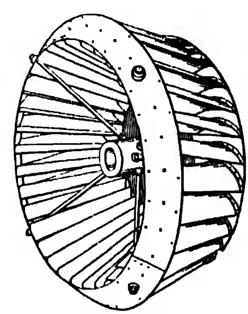


FIG. 231. Single Conoidal Fan Wheel.

generally known as **multi-vane** fans. The former are primarily intended for slow-speed drives and the latter for direct connection to high-speed motors or steam turbines. The blast wheel of the steel-plate fan has 5 to 12 radial blades, depending upon the size of the fan. The blades are of heavy steel plate riveted to cast-steel or structural-steel spider arms. The housing is involute in form and made of heavy steel plates, the scroll being of such proportions that the velocity is gradually reduced without

loss or shock. The inlet cone is designed to give a gradual increase of velocity with a minimum loss. The blast wheel of the multi-vane fan has 20 to 60 short pressed-steel blades riveted to the back and front plates as shown in Fig. 230. For high velocities, in order to withstand the centrifugal stresses, the blades are frequently split up and reinforced by annular stiffening rings, as illustrated in Fig. 232. The multi-vane fan has practically supplanted the steel-plate type in the large modern power house, because of its compactness for a given capacity. The tendency toward high rotative speed in mechanical-draft apparatus is evidenced by the helicoidal impeller-type of runner manufactured by the Rateau Battu Smoot Company. In one design the impellers are 18 in. in diameter and operate at 5000 r.p.m., giving a tip speed of 23,500 ft. per min., and a corresponding static pressure of 7 in. of water when delivering 30,000 cu. ft. of free air per min. Fans may also be classified with respect to the direction of the blade at the periphery, as: (1) the forward curve, (2) the radial tip, (3) the partial backward curve, and (4) the full backward curve. Each shape influences the relation between pressure, efficiency, power requirements, speed, and capacity, and controls the selection for a given set of operating conditions. The housings for the rotor may be arranged for top or bottom horizontal discharge, up or down blast, or any other position depending upon the arrangement of the draft system.

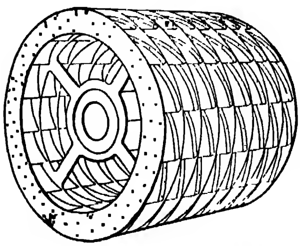


Fig. 232. High-speed, Multi-vane Fan Wheel.

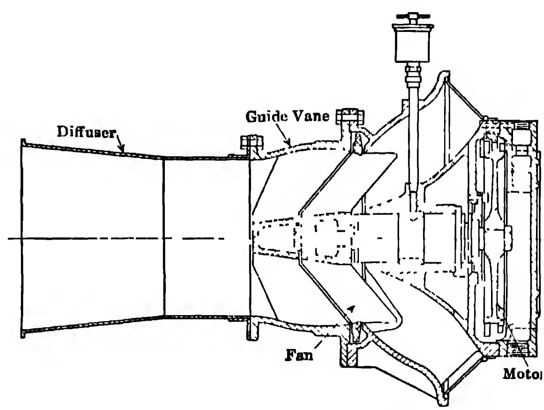


Fig. 233. Sectional View of Turbine-driven "Vano" Blower.

Figure 233 shows a section through a **Coppus** turbine-driven **"Vano" blower** which differs considerably from the types previously described both in principle and in operation. The blower is a screw-blade propeller so designed that the air leaves the blades in the same direction as it enters. The air leaving the propeller is forced through guide-vane blades which have a curvature increasing in the direction of rotation of the propeller,

and which, in conjunction with a diffusing cone, convert a considerable part of the velocity pressure to static. The particular blower illustrated is driven by a small self-contained steam turbine, but direct-connected motors may be used in place of the turbines. Vano blowers operate against pressures up to 8 in. of water, and the manufacturers claim efficiencies up to 80 per cent with practically no variation in power consumption at constant speed for variations in air delivery or pressure. See Fig. 241.

157. Elementary Theory of Fans. — The advent of the underfeed stoker, calling for large volumes at higher pressures than had been previously required from fans, necessitated stronger and heavier construction for slow-speed engine drives and basically new designs for high-speed motor and turbine drives. The fundamental theory of air flow is the same for all types of centrifugal blowers, but the actual performance is dependent upon so many variables involving constructive details that general rules for design are without purpose. The development of a particular type of fan is largely a matter of experiment, and the design of blade shapes, blade angles, and the like is based primarily upon the results of these tests. Because of the vast amount of data involved, no attempt will be made to analyze the problem of design, and only such elementary theory will be discussed as is necessary for a clear understanding of the principles of operation.

Pressure. — The main object of a forced-draft fan in a bulk-fuel-fired plant is to force air through the fuel bed in quantities sufficient for combustion and under pressures high enough to overcome the various frictional resistances. In forced-draft powdered-fuel-burning plants, the fuel is carried in suspension in the combustion air, but, because of the frictional resistances in the burner equipment and the high velocity of the jet, high initial static pressures are frequently necessary. Similarly, the induced-draft fan must be capable of maintaining a slight suction over the fire under all conditions of load. Air or gas in motion in a conveying system has three distinct pressures, namely, velocity, static, and dynamic. **Velocity pressure,** as the name implies, is the head required to impart motion to the fluid; **static pressure** is the head required to overcome the resistance offered to the flow; and **dynamic pressure** is the sum of the static and velocity heads. Since the resistance to flow is large in average mechanical-draft practice, it is evident that the greater the static pressure developed at the fan outlet in proportion to the velocity pressure, the more effective will be the performance of the fan.

If the delivery or suction pipe of a fan is sealed against flow, there can be but one pressure, namely, static. Referring to Fig. 234 " *A* " represents a manometer-equipped Pitot tube inserted in the suction or discharge

conduit of a centrifugal fan so as to face the current, and " *B* " is a manometer attached to an opening in the wall of the casing. For accurate determinations manometer " *B* " is attached to a piezometer ring. "*A* " receives the full impulse of the stream and the manometer indicates the total or dynamic pressure, while "*B* " registers the static pressure only. With the pipe sealed against discharge, resistance to flow is a maximum, there is no flow and the liquid depression in both manometers is the same; therefore, the static and dynamic pressures are equal, or, what amounts to the same thing, there is only static pressure in the conduit. If the conduit is opened to its maximum and there are no frictional resistances, the static pressure indicated by manometer "*B*," Fig. 235, becomes zero while that in " *A* " stands at a height equivalent to the full impulse of the stream, that is, there is only velocity pressure in the conduit. The liquid depression in manometer " *A* " is a measure of the velocity of the air at the point where the mouth of the Pitot tube is located. Since the velocity is greater at the center than near the walls of the conduit, it is necessary to take a large number of readings at various points in order to obtain the average velocity. The velocity of air at a given density which will give a manometer depression of one inch of water is known as the **velocity constant** of air at that density.

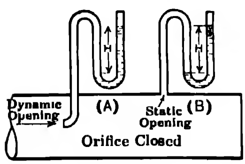


FIG. 234

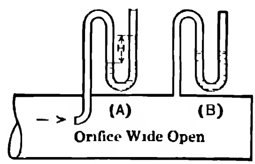


Fig. 235

If the flow is restricted as by throttling, there is a depression in both manometers, Fig. 236, that is, there is both velocity and static pressure in the conduit. The difference between the depressions in " *A* " and " *B* " is the head due to velocity. By connecting the two manometers as outlined in Fig. 236 (*C*), the velocity pressure is given directly.

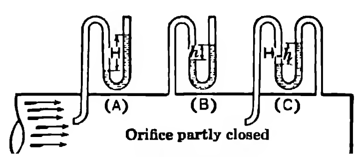


Fig. 236

Pressure resulting from a current of air or gas flowing at a velocity corresponding to that of the tip of the blades is designated by fan builders as **peripheral-velocity pressure.**

In any centrifugal fan, the pressure is the resultant of centrifugal force due to the rotation of the air within the wheel and the kinetic energy contained in the air by virtue of its velocity upon leaving the tips of the vanes or blades. In mechanical-draft practice, the kinetic energy of the air leaving the periphery must be converted largely into potential energy in the form of static pressure before being serviceable.

This conversion is ordinarily effected in the scroll formation of the fan housing.

The lowest pressure required of a forced-draft fan is usually about 2 in. static pressure, which occurs in hand-fired practice, and the highest draft necessary at 300 to 400 per cent of boiler rating is approximately 7 in. static pressure, where underfeed stokers are used. Static pressures for induced-draft fans range from 1.5 in. to 7 in. depending upon the resistances to be overcome and the temperature of the flue gases.

For a given fan-piping system and air density, the pressure developed varies as the square of the speed, but because of the numerous factors involved, such as blade shapes, blade angles, housing designs, capacity, and the like, exact values cannot be expressed by simple mathematical equations and recourse must be had to **characteristic curves** plotted from actual tests. (See paragraph 158.)

Velocity and Pressure. — In all centrifugal fans, the velocity of the air leaving the blades bears a definite relation to the peripheral velocity of the fan wheel. This relationship is greatly influenced by the design of the blading, as will be seen from an inspection of Fig. 237. The line u

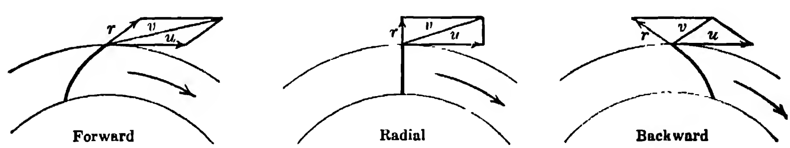


Fig. 237

represents the peripheral velocity in direction and amount; r the velocity of the centrifugal force of the air, and v the resultant velocity with respect to the fan casing. It will be seen that with the radial and bent-forward blade the resultant is greater than the peripheral velocity, while with the blades bent backward the resultant velocity is less. By changing the direction of curvature a wide range in resultant velocities is possible. The velocity of the air leaving the tips of the blades is greatly in excess of that ordinarily required in mechanical-draft systems. By enclosing the wheel in a casing having a properly designed scroll, part of the velocity pressure is converted to static. The velocity at the outlet of the average fan at rated capacity is approximately one-half the peripheral velocity. For ordinary fan work, where the air or gas is but slightly compressed, the relationship between pressure and velocity is substantially

$$V = \sqrt{2gh} \tag{112}$$

in which

V = velocity of flow, ft. per sec.,
g = acceleration of gravity, 32.2 (approx.),
h = head of gas causing flow, ft.

By converting "h" in feet of gas to the equivalent pressure in inches of water, and considering time in minutes instead of seconds, equation (112) may be expressed

$$v = C\sqrt{p \div d} \qquad (113)$$

in which

v = velocity, ft. per min.,
C = constant = 1096,
p = pressure drop producing velocity, in. of water at 62 deg. fahr.,
d = density of the gas, lb. per cu. ft.

For standard conditions, dry air at 70 deg. fahr., barometer 29.92 in., relative humidity 70 per cent, d = 0.07465* hence

$$v = K\sqrt{p} = 4011\sqrt{p} \qquad (114)$$

Where quietness of operation is necessary, the velocity in the ducts should be limited to 2000 ft. per min., but where this is not essential, velocities as high as 5000 ft. per min. may be used. This refers only to cold-air systems. For hot gases, as in connection with induced-draft fans, the velocity may be practically doubled. The friction losses increase rapidly with the velocity, so that the usual compromise must be made between size and velocity; otherwise, the pressure drops become excessive.

Capacity. — For a given fan size, piping system, and air density, the capacity, Q, varies directly as the velocity and hence as the speed of the fan, thus,

$$Q = vA \qquad (115)$$

in which

Q = volume, cu. ft. per min.,
v = velocity, ft. per min.,
A = area of the conduit, sq. ft.

Since the velocity varies as the square root of the pressure drop

$$Q = KA\sqrt{p}, \qquad (116)$$

* A.S.H.V.E. and N.F.A. Code. A.S.M.E. Code recommends 68 degrees temperature and density of 0.075.

in which

K = coefficient determined by experiment; other notations as in equations (113) and (115).

Horsepower. — The horsepower required to operate a fan varies directly with the capacity and the total or dynamic pressure, thus:

$$\text{Hp.} = \frac{5.2\,Q \times P_d}{33{,}000 \times E} = 0.000157\,\frac{Q \times P_d}{E} \tag{117}$$

in which

E = total efficiency of the blower,
P_d = dynamic pressure, in. of water.

Combining equations (116) and (117) and reducing, remembering that for constant orifice conditions and at known air density the velocity pressure bears a definite relation to the peripheral velocity, we have

$$\text{Hp.} = Bp^{\frac{3}{2}} \tag{118}$$

in which

B = coefficient involving all constants and reduction factors.

Equation (118) shows that the horsepower varies as the cube of the square root of the pressure.

Since the capacity is directly proportionate to the peripheral velocity or fan speed, and the pressure developed varies directly as the square of the speed, it follows that the horsepower varies as the cube of the speed, thus:

$$\text{Hp.} = MN^3, \tag{119}$$

in which

M = coefficient involving all constant and reduction factors,
N = speed of the fan, r.p.m.

The marked increase in power for even a moderate increase in speed should be borne in mind in selecting a fan. It is, as a rule, more economical to err in selecting too large a fan than one which must be forced above its rated speed. The capacity varies directly with the speed; therefore, the horsepower also varies with the cube of the capacity.

Equations (115) to (119) are based on the assumption that the resistance is constant. In underfeed-stoker practice the resistance is not constant, and therefore the fans do not follow these laws. The dotted curves in Fig. 238 show the relation between volume and pressure in accordance with the constant-resistance law, while the full-line curves show the actual

relationship for a specific case. Figure 239 shows the relation between draft pressure and rate of driving for an underfeed-stoker equipment in which there is a decided falling off in pressure requirements at the higher

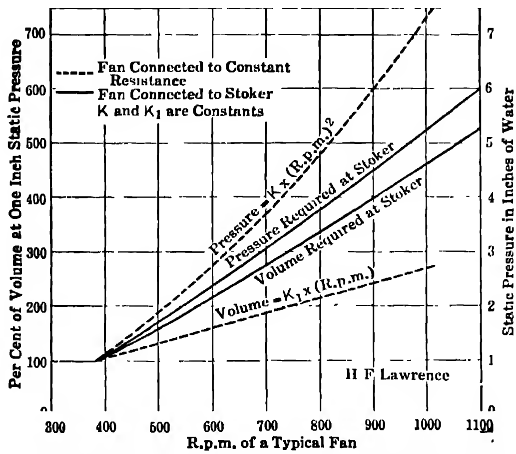


FIG. 238. Relation of Volume and Pressure of Air to Fan Speed.

ratings. This is due to the extreme agitation of the fuel bed at heavy loads. Table 41 shows the influence of the solid constituents on the resistance through fuel bed and grate with underfeed stokers.

Manometric Efficiency. — This efficiency is the ratio of the dynamic head as actually observed to the maximum theoretical dynamic head, or

$$E_{\text{man}} = h/H = gh/u^2 \qquad (120)$$

in which

h = actual dynamic head, ft. of fluid,
g = acceleration of gravity = 32.2,
u = tip speed, ft. per sec.

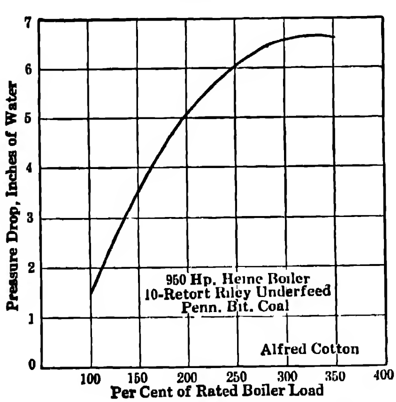


FIG. 239. Curve Showing Pressure Drop through Fuel Bed. Underfeed Stoker.

Volumetric Efficiency. — This is ..ally not an efficiency and might better be called volumetric capacity. It is defined as the actual volume of air passing in a given time divided by the impeller displacement for the same period, or

$$E_{\text{vol.}} = 4Q/\pi D^2 NB \qquad (121)$$

in which

Q = volume discharge, cu. ft. per min.,
D = diameter of the impeller, ft.,
B = width of the impeller, ft.,
N = r.p.m.

Mechanical efficiency (Standard Code of Fan Testing, A.S.H.V. and N.A.F.M.) is the ratio of the total work done by the fan in moving the air to the hp. input of the fan, or

$$E_{\text{mec.}} = Qh \div H_i \times 33{,}000 \qquad (122)$$

in which

Q = weight discharged, lb. per min.,
h = dynamic head, ft. of air,
H_i = hp. input.

Two efficiencies are sometimes given, (1) that based on the dynamic head as in equation (122) and (2) that based on the static head. The former is more generally used. According to the Standard Code of Fan Testing as recommended by the joint committee of the American Society of Heating and Ventilating Engineers and the National Association of Fan Manufacturers, the friction head of ducts shall be determined from the following rules:

$$\text{For round ducts, } f = 0.0257Lh/D \qquad (123)$$

$$\text{For square or rectangular ducts, } f = 0.01285Lh\,(a + b) \div ab \qquad (124)$$

in which

f = pressure drop due to friction, in. of water,
L = distance from fan outlet to point in duct where measurements are made, ft.,
D = diameter of the duct, ft.,
a = long side of the duct, ft.,
b = short side of the duct, ft.,
h = velocity pressure, in. of water.

The factor for elbows is difficult to determine with any degree of accuracy, but for rough approximations the pressure drop for one right-angle bend may be taken as that due to a duct 10 diameters in length.

Standard Code for the Testing of Centrifugal and Disc Fans: Jour. Am. Soc. H. & V. Engrs., May, 1923, p. 371.

Fan Blower Design: Jour. Am. Soc. H. & V. Engrs., July 22, 1922, p. 491.

Pitot Tube for Gas Measurement: W. C. Rowse, Trans. A S.M.E., Vol. 35, 1913, p. 633.
The Impact Tube: S. A. Moss. Trans. A.S.M.E., Vol. 39, 1916, p. 761.
Some Development in Centrifugal Fan Design: National Engr., Jan., 1922, p. 6.

158. Performance of Fans. — There are so many different types of fans on the market, and the variable factors involved in their operation are so

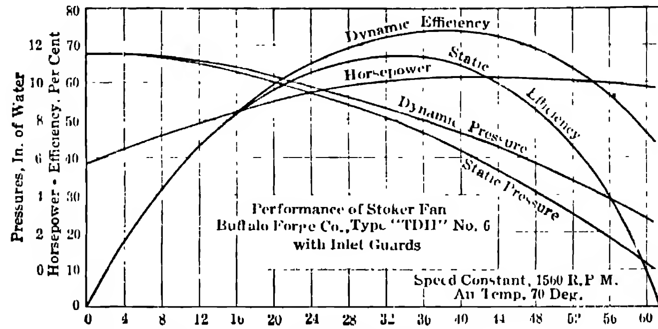


FIG. 240. Performance of Stoker Fan. (Constant Speed.)

numerous, that attempts to analyze performance mathematically are without purpose. For this reason, fan manufacturers furnish **capacity tables** and **characteristic curves,** based on actual tests, which give the

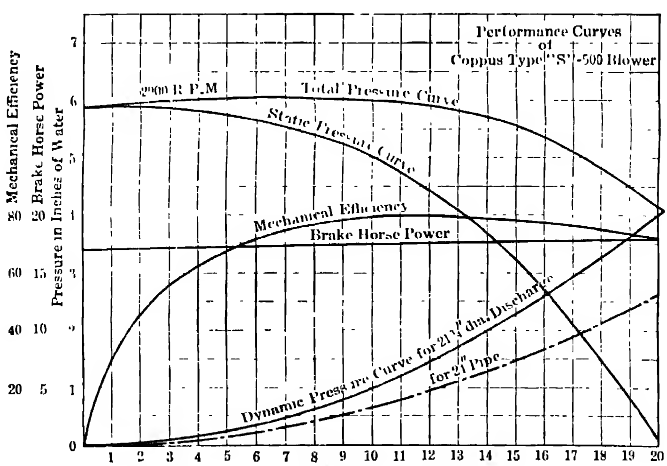


FIG. 241. Performance of Coppus Type "S" Vano Blower.
(Constant Speed.)

performance of their product over a wide range of operation. With the aid of these tables and curves, the proper size and type of fan for a given set of operating conditions may be selected with intelligence.

Two sets of capacity tables are to be had from manufacturers, viz., **rated capacity,** Table 53, and **variable capacity,** Table 54. The former gives the capacity, speed, and hp. of the different fans for various static and total pressures when operating at what is approximately the highest

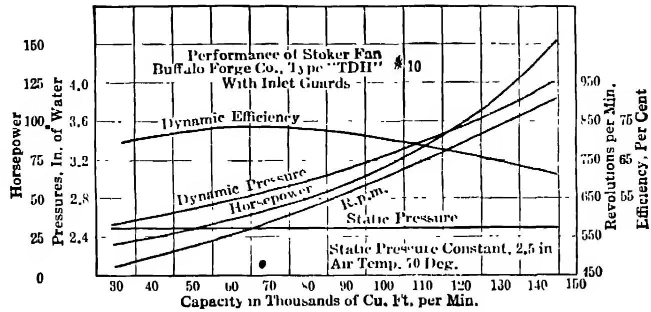


FIG. 242. Performance of Stoker Fan. (Constant Static Pressure.)

efficiency. The variable capacity tables give the performance of each size of fan on either side of the condition for maximum efficiency.

Characteristic curves are graphical charts visualizing the relationship between capacity, speed, pressures, horsepower, and efficiency. They are plotted in a variety of forms, a few of which will be briefly discussed.

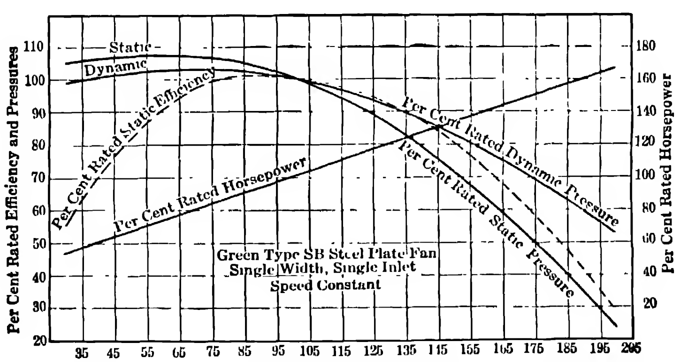


FIG. 243. Performance of Steel-plate Fan. Standard Characteristic Curves.

Figures 240–242 give the actual test results of a specific type and size of fan under variable conditions of speed, static pressure, and capacity. These curves are readily interpreted because the various quantities may be read directly from the chart. They are limited, however, to the performance of the particular size of fan tested. The characteristics for all sizes of any given design of fan are practically the same; therefore, if the coördinates are expressed in terms of "percentages" of the performance

at rated capacity, one set of curves will suffice for all sizes. This method of representing the curves is recommended by the Joint Committee of the American Society of Heating and Ventilating Engineers and the National

TABLE 53

TYPICAL "RATED CAPACITY" TABLE

CAPACITIES OF FORCED-DRAFT FANS

Green "Radial Flow"

(Multi vane Type)

Diam. Wheel In	Performance	Static Pressure at Fan Discharge, In. of Water *						
		1	2	3	4	5	6	7
20½	Cu. Ft.	2,800	4,000	5,200	6,000	6,400	7,200	7,600
	R.p.m.	860	1,216	1,505	•1,738	1,927	2 122	2,280
	Br. hp.	0 62	1 77	3 49	5 36	7.15	9.65	11.85
24	Cu. Ft.	4,000	5,500	7,000	8,000	9,000	9,500	10,500
	R.p.m.	741	1,013	1,284	1,481	1,658	1,805	1,951
	Br. hp.	0.87	2 17	4 69	7 15	10.0	12 8	16.4
27¾	Cu. Ft.	5,600	7,700	9,100	10,500	11,900	12,600	14,000
	R.p.m.	648	913	1,110	1,280	1,435	1,563	1,696
	Br. hp.	1.22	3 41	6.11	9.39	13.4	17.6	21 9
32¼	Cu. Ft.	7,200	9,900	12,600	14,400	16,200	17,100	18,900
	R.p.m.	550	776	955	1,102	1,232	1,342	1,456
	Br. hp.	1.57	4 41	8.46	12.9	18.1	22.9	29.5
37½	Cu. Ft.	10,000	13,750	16,250	18,750	21,250	23,750	25,000
	R.p.m.	478	671	816	944	1,055	1,161	1,248
	Br. hp.	2.18	6.13	10.9	16.8	23.8	31 8	39.0
43	Cu. Ft.	13,500	18,000	22,500	25,500	28,500	31,500	33,000
	R.p.m.	416	583	714	821	917	1,007	1,083
	Br. hp.	2.97	8.06	15.1	22.8	31.9	42.3	51.6
50	Cu. Ft.	18,000	24,000	30,000	36,000	38,000	42,000	46,000
	R.p.m.	351	495	609	707	782	859	930
	Br. hp.	3 95	10.7	20 1	32.3	42.5	56.3	71.9
58	Cu. Ft.	22,000	34,000	40,000	46,000	52,000	58,000	61,000
	R.p.m.	301	433	525	606	678	746	803
	Br. hp.	4.86	15 2	26 9	41.3	58.0	77.8	95.9
67¼	Cu. Ft.	30,000	46,000	51,000	62,000	70,000	78,000	82,000
	R.p.m.	260	372	452	521	583	641	689
	Br. hp.	6 57	20 5	36 1	55.6	78.1	104	12.8
78	Cu. Ft.	45,000	60,000	75,000	85,000	95,000	105,000	110,000
	R.p.m.	228	319	391	451	504	553	594
	Br. hp.	10 2	26.8	50 2	76.0	107	141	172
90½	Cu. Ft.	57,500	80,000	102,500	117,500	125,000	140,000	155,000
	R.p.m.	194	274	337	389	431	474	515
	Br. hp.	12.7	35.3	68.5	105	140	187	242

Cu. Ft. per Min. at 70 deg. fahr. and at point of maximum efficiency.

Association of Fan Makers. Figures 243 and 244 are illustrative of this class. Referring to Fig. 243, if the fan is operated at, say, 70 per cent of rated capacity, the horsepower, dynamic pressure, static pressure, and efficiency will be 60, 103, 107, and 98 per cent, respectively, of that at rated capacity.

In calculating the performance of fans of the same design and similar proportions, but of other sizes and at other speeds, the following law applies: Up to and within 1 1/2 lb. per sq. in. pressure difference, and at the same peripheral speed and discharge conditions, the delivery varies as the square of the diameter of the wheel, or, for different speeds, the delivery varies as the cube of the diameter times the number of r.p.m.

In correcting the readings of a given fan for constant discharge conditions, the following law applies: Capacity varies directly as the speed; pressures as the square of the speed; and horsepower as the speed cubed.

TABLE 54

TYPICAL "VARIABLE CAPACITY" TABLE

Performance of Buffalo Forge Co.'s No. 6 Duplex Conoidal (Type DDH)

Outlet Velocity Ft. per Min.	Capacity Cu. Ft. Air per Min.	Add for Total Pressure	Static Pressure, In. of Water. Barometer 29.92, Temperature 70 Deg. Fahr.													
			4		5		6		7		7½		8		8½	
			RPM	HP	RPM	HP	RPM	HP	RPM	HP	RPM	HP	RPM	HP	RPM	HP
1600	16,800	160	833	19 0	927	25 1	1014	31 5	1092	38 6	1130	41 7	1166	45 8	1201	49 3
1800	18,895	202	837	20 5	931	26 7	1018	33 1	1096	40 6	1133	44 3	1169	47 7	1204	51 9
2000	21,000	250	841	22 3	935	28 5	1021	35 4	1100	42 7	1137	46 5	1173	50 6	1208	54 2
2200	23,092	302	848	24 1	940	30 6	1025	37 6	1104	45 1	1140	49 1	1177	53 1	1212	56 8
2400	25,190	360	855	26 2	946	32 8	1030	40 2	1109	47 7	1144	51 7	1181	55 7	1216	59 8
2600	27,295	422	864	28 4	952	35 4	1035	42 8	1114	50 6	1150	54 6	1185	58 8	1221	63 0
2800	29,395	489	872	30 9	960	38 2	1042	45 9	1119	54 0	1156	58 1	1190	62 2	1225	66 4
3000	31,400	560	883	33 4	968	41 1	1050	49 1	1124	57 4	1162	61 6	1195	66 2	1230	70 1
3200	33,590	638	895	36 4	980	44 3	1058	52 6	1131	61 1	1169	65 3	1200	70 2	1235	74 3
3400	35,690	721	908	39 4	991	47 7	1068	56 2	1141	65 1	1175	69 6	1208	74 2	1242	78 6
3600	37,785	808	922	42 5	1003	51 2	1078	59 8	1150	69 2	1182	73 6	1216	78 3	1251	83 3
3800	39,885	890	938	46 1	1016	55 2	1089	63 9	1160	73 6	1192	78 4	1225	83 5	1260	88 0
4000	41,990	995	955	49 7	1030	50 0	1101	68 5	1171	78 3	1201	83 4	1235	88 7	1271	93 4
4200	44,080	1 098	970	53 8	1045	63 3	1115	73 1	1182	83 5	1217	88 4	1246	93 3	1282	98 8
4400	46,190	1 204	988	58 2	1061	67 8	1130	78 2	1196	88 7	1229	93 6	1258	99 5	1292	105 0
4600	48,285	1 317	1005	62 7	1077	73 2	1145	83 5	1209	94 0	1240	99 5	1270	105 4	1303	111 1
4800	50,390	1 432	1023	67 9	1095	78 2	1160	88 6	1221	100 2	1252	105 5	1282	111 9	1315	116 9
5000	52,490	1 552	1045	72 5	1111	83 5	1176	94 5	1237	106 0	1268	112 0	1296	118 1	1328	123 5
5200	54,585	1 679	1065	77 2	1130	89 2	1192	100 9	1254	112 5	1283	118 5	1312	124 3	1341	130 4
5400	56,690	1 812	1083	83 5	1150	95 0	1210	107 0	1270	119 2	1290	125 3	1328	131 2	1358	138 0
5600	58,790	1 950	1105	80 2	1169	101 2	1230	114 0	1287	126 8	1315	132 1	1345	138 7	1375	145 5

159. Selection of Type and Size of Fan. — The influencing factors in the choice of a fan for mechanical-draft purposes are primarily the volume of air or gas to be handled, static pressure necessary to overcome the frictional resistance of the system, and the horsepower to drive the fan. Other factors which may be of equal or even greater importance are reliability, successful parallel operation, high static efficiency, and a reserve of pressure for variation in load. Air and draft pressure requirements at various loads may be approximated, as outlined in previous chapters. The next step is to select from fan-capacity tables the different sizes and types of fans which will deliver the desired maximum volume of air at the required maximum static pressure. Care must be taken to include in the static pressure the various drops due to the resistances of the air ducts. It will be noted that several types and sizes of fans will give the required volumes and pressures. The list may be greatly reduced by eliminating the sizes which range beyond the desired speed and for which the power requirements are excessively high. Thus, for low rotative speeds, the steel-plate fan will probably be the best investment, and for high speeds some type of multi-vane blower is to be preferred. Since the horsepower for a given capacity and static head runs up rapidly with the speed, the size consuming the least power for the average boiler load is to be given preference to the others, though first cost must also be considered.

The next step is to obtain from the manufacturer characteristic curves or variable capacity tables for the particular types selected, and compare the static pressures at various capacities of the fan with the calculated pressures and volume requirements. For a constant speed drive, the curves will be of the form shown in Fig. 240, and for variable speed on the order of those shown in Fig. 242. Considering the different characteristics in order of the curvature of the blade tip, viz., full-forward, radial, partial-backward, and full-backward, we have:

Full-forward Tip. — This is virtually a velocity fan, since the rotational component of the air velocity leaving the wheel is actually higher than the rotational speed of the wheel itself. This feature is of advantage where very high outlet velocity is required and noise is not objectionable. It has the slowest tip speed, for a given pressure, of any type. The rising pressure curve (*a*) of Fig. 244 from full maximum to the load corresponding to maximum efficiency is desirable in that the pressure builds up without change in speed should the volume of air decrease, as when the resistance through the fuel bed is increased by clinker formation. The flat or drooping pressure curve, between the point of maximum efficiency and practically zero capacity, is undesirable for parallel operation, since there is no assurance that one of the fans will not "lie down" if the flow of air to the inlet of the two fans is not equally unobstructed. Furthermore, any

change in relative speed will increase the unbalancing effect. The upward hp. curve with increase in capacity represents a constant danger of overload on the driving motor, necessitating an oversized drive to take care of the maximum requirements. Because of the undesirable characteristics, the full-forward tip fan is little used for forced-draft service. Where both forced- and induced-draft fans are used, the resistances for the induced-draft fan are practically constant; the exact form of the pressure characteristic for the latter is therefore unimportant and the speed element becomes the deciding factor. High-speed induced-draft fans are undesirable because the passages between the blades are small and a deposit of soot or dust impairs the efficiency and capacity. Therefore, if the multiblade high-speed type is selected for this service, the forwardly inclined tip offers the advantage of maximum pressure at the lowest peripheral speed. Induced-draft fans in many of the ultra-modern power stations are of the forwardly inclined type.

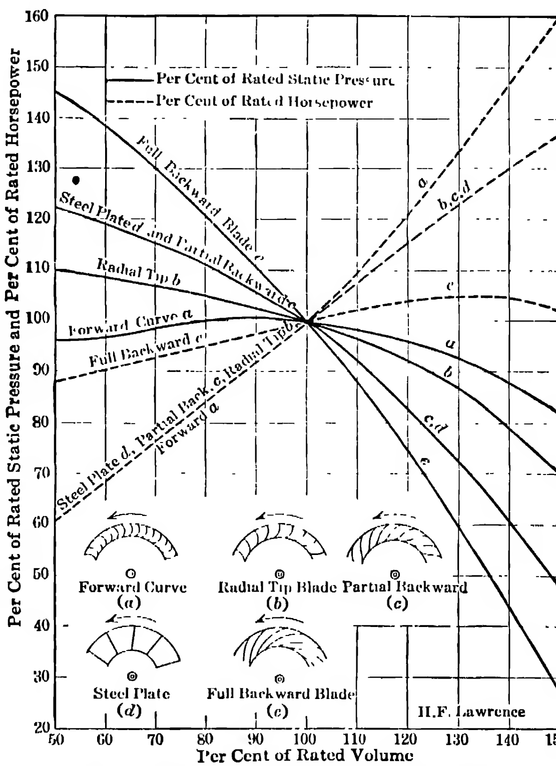


FIG. 244. Fan Characteristics at Constant Speeds.

Radial Tip. — The pressure characteristics of this type, Fig. 244 (*b*), show that the tendency is toward maintaining constant pressure over a wide range in capacity. The flat portion of the curve makes the fan very sensitive to resistance variation, and, if used at a capacity corresponding to this portion of the curve, may cause the fan to run under or over the estimated capacity. This rising hp. curve is also undesirable.

Full-backward Tip. — In this design, Fig. 244 (*e*), the pressure curve rises with decrease in capacity from the maximum capacity obtainable to zero, which enables the fan without change of speed to overcome any

variation in resistance of the fuel bed. This rising characteristic also permits of perfect parallel operation. It will be noted that the hp. curve rises slowly to a certain capacity and then drops off. This self-limitation in power requirements is ideal for motor drive, since there is no danger from overloading. The efficiency is high and the fan has the highest tip speed, for a given pressure, of any type. These characteristics are all favorable for high-speed forced-draft service and the majority of the modern installations are equipped with fans of this type.

Partial-backward Tip. — This is an intermediate design between the full-forward and full-backward type. The blades are curved forward at the heel to meet the incoming air and backward at the tip to discharge it, Fig. 244 (*c*). By changing the inclination and curvature of the blades, practically any characteristic from that of the full-forward to that of the full-backward design is obtainable. The pressure characteristic for the particular design shown in Fig. 244 (*c*) is favorable for forced-draft service, but the hp. characteristic is not self-limiting. The self-limiting hp. feature, however, may be developed by properly proportioning the vanes. With variable-speed drive and automatic control, the self-limiting power curve is of secondary importance.

Steel-plate Fans. — The pressure and power characteristics of the steel-plate or few-bladed paddle-wheel fans, Fig. 244 (*d*), are practically identical with those of the double-curved blades illustrated in Fig. 244 (*c*). The pressure curves are suitable for forced- or induced-draft, but the hp. curve is not self-limiting. Steel-plate fans are inherently slow-speed devices and are usually installed where the driving units are steam engines. The slow-speed feature is desirable in induced-draft installations where cinders must be handled.

Where the character of the station load curve is known or where it can be approximated with a fair degree of accuracy, it is not a difficult matter to select a size and type of fan which conforms with the station requirements; but such knowledge is ordinarily the exception and the choice is dependent largely upon experience.

Plotting Blower Test Curves: A. H. Anderson, Trans. A.S.M.E., Vol. 39, 1917, p. 793.

160. Mechanical-draft Fan Drives. — While a large number of forced- and induced-draft fans in the older plants are of the slow-speed paddle-wheel types, driven by small vertical engines, and many of the modern plants employing high-speed multi-vane fans are equipped with direct connected or geared turbines, the modern tendency is toward motor drive. Induction motors of the slip-ring type are the more common in the latest designed plants, but several plants are equipped with variable-speed direct-current and variable-speed alternating-current motors with brush-

shifting mechanisms. Automatic control is commonly used with steam-turbine drives and hand control with motors, though automatic-motor control is finding favor with many engineers. Alternating-current motors of the induction type seem to be preferred for induced-draft fans. Variable-speed direct-current motors are also used for induced-draft service. Practice is divided between the use of individual fans for each boiler and the common duct with several fans discharging into it. Some idea of current practice in the selection of a fan drive may be gained from the data in Table 55.

TABLE 55

FAN DRIVES IN MODERN CENTRAL STATIONS

Station	Forced Draft	Induced Draft
Cahokia	2300-volt, a.c. constant speed	None
Calumet	440-volt induction motor	2300-volt squirrel-cage
Colfax	Geared turbine	None
Delaware	Induction motor	Induction motor
Hell Gate	2200-volt, a.c. brush-shifting motor	Same as forced draft
Marysville	Variable speed, 240-volt d.c.	Variable speed, 240-volt d.c.
Seward	Geared turbine	None
South Meadow	Geared turbine	None
Springdale	2300-volt, a.c. brush-shifting motor	Same as forced draft
Weymouth	2300-volt, a.c. brush-shifting motor	Same as forced draft
Windsor	500-volt induction motor	Same as forced draft

Control for Power Station Auxiliary Motors: Power Plant Engrg., June 1, 1923, p. 581. *Driving Power-House Auxiliaries:* Power, Jan. 31, 1922, p. 166; May 20, 1924, p. 817.

161. Fan-draft Control. — The volume of air for combustion may be varied to meet the steam demand by throttling where the fans operate at constant speed, and by changing the speed of rotation where variable-speed drives are employed. The air gates or dampers and the speed of the fans may be manually or automatically controlled. Manual control may be effected at the point where the air gates, dampers, or auxiliary drives are located or, from distant points, (remote control) through the agency of suitable relay apparatus. With automatic control it is customary to coördinate the manipulation of the air-supply apparatus with that of the stack damper in case of hand firing, and with that of the stoker drive in case of stoker firing. The primary controlling forces are variation in (1) steam pressure, (2) furnace suction and (3) air-duct pressure, separately or in combination with each other. These forces actuate suitable relay mechanisms which in turn vary the positions of the air gates or dampers, and the speeds of the stoker and fan drives. Among the popular makes of combustion-control systems may be mentioned the **Balanced**

Draft, Hagan, Merritt, Hess-Benjamin, Ruggles-Klingemann, and **Smoot.**

In the **Balanced Draft** system of the Engineer Company, each boiler unit is individually controlled. The speed of the fan is controlled by a diaphragm regulator actuated by variations in plant header pressure. Movement of the diaphragm is transmitted to a pilot valve, which in turn admits water under pressure or compressed air to a piston. The piston movement changes the position of the controlling mechanism on the fan drive and in this manner varies the speed. The stack damper is open or closed by a " furnace-pressure regulator " which consists essentially

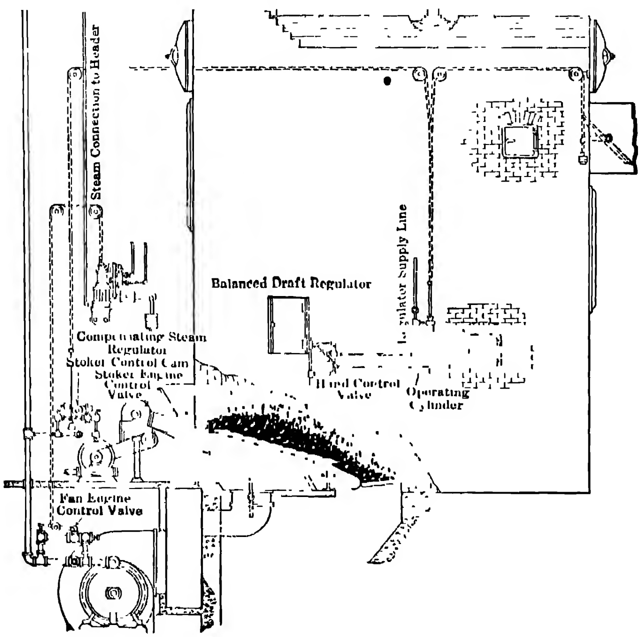


FIG. 215. "Balanced Draft" Combustion Control.

of a swinging blade diaphragm actuated by draft-pressure variation in the furnace. Movements of the diaphragm operate a piston through the agency of a pilot valve, in much the same manner as the fan control. This piston operates the damper. The speed of the stoker drive is varied by the same piston which operates the flue damper. The movement of the piston, however, is not transmitted directly to the stoker speed control but indirectly through an adjustable cam. This system is first adjusted manually to meet the specific requirements of the particular plant to which it is applied, and the various adjustments are synchronized to give the best results. After this adjustment, further control is automatic.

The operation is as follows: If there is an increase in load on the plant, there will be a pressure drop in the steam header. The steam regulator will then gradually speed up the fan, force more air through the fuel bed, and increase the rate of combustion. This will produce more gas and tend to decrease the draft in the furnace chamber. The furnace-pressure regulator will immediately increase the damper opening just enough to remove the gas at the new rate and maintain the predetermined furnace draft. At the same time, the stoker-control cam will be turned to a new position and the correct amount of fuel to support the increased rate of

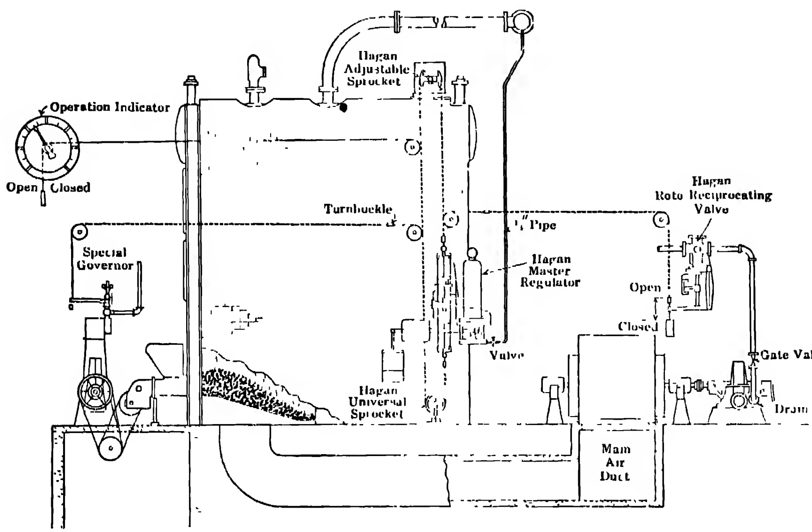


FIG. 246. Hagan Combustion Control.

combustion will be fed. In case the forced-draft fan is driven by a constant-speed motor or turbine, the cam mechanism varies the position of the blast gate.

In the **Hagan Combustion Control** system, the primary controlling force is the difference in pressure across an orifice plate inserted between the flanges in the main steam header. The pressure difference at the orifice is carried to a balanced piston (master regulator) the movement of which is transmitted through chains and pulleys to the damper, the stoker and fan drive. The adjustments are such that movements of the boiler damper and speed of fan and stoker drives are synchronized to maintain the predetermined relation between fuel and air supply and to vary the rate of fuel and air supply in accordance with steam demand.

The **Hess-Benjamin** system regulates direct-current motor-driven stoker and fan speeds by a master regulator consisting essentially of a properly

designed exciter for the shunt fields of all motors. The exciter current is varied by means of a balanced piston, the movement of which is actuated by a pressure drop in the main steam header from points on the saturated steam nozzle of the boiler to the points just ahead of the prime movers.

Combustion Control for Steam Boilers: Power, Mar. 6, 1923, p. 354; May 13, 1924, p. 761; Feb 24, 1925, p. 311; Mech Engrg., Mar. 1925, p. 193.

Centralized Combustion Control for Boilers: Power, May 16, 1922, p 771

Fuel Saving Effected by Combustion Control Power Plant Engrg., May 1, 1923, p. 473.

Automatic Combustion Control: Power Plant Engrg., July 1, 1920, p. 649; July 1, 1924, p. 694.

Electrical System of Combustion Control for Boiler Plants: Power, Jan. 8, 1924, p. 46.

PROBLEMS

1. The over-fire air supply of a 100-hp. horizontal return-tubular boiler when operating at rating is furnished by two 3/16-in. diameter steam jets, steam pressure at nozzle 65 lb per sq in gage What percentage of the weight of steam generated by the boiler is required to operate the jets?

2. Dry air is flowing through a conduit, the velocity head (as indicated in Fig 236) being 1 in. water. If 1 cu. ft. air weighs 0.071 lb, required the velocity in ft. per sec.

3. Let the cross-sectional area of the conduit in Problem 1 be 2 sq ft and the static pressure 0 5 in water. Required the output horsepower of the fan.

4. It is required to supply 20,000 cu. ft. air per min. to a furnace under a static pressure of 1 in water. The conduit is 10 ft. square and 100 ft. long. Static pressure drop due to right-angle bends 0.1 in. The mechanical efficiency of the fan is 60 per cent. One cu. ft. of air weighs 0.074 lb. Calculate the horsepower required to drive the fan.

5. Required the horsepower necessary to operate the fan in Problem 3 if the static pressure is increased to 2 in., other conditions remaining the same.

6. If the rated speed of fan in Problem 3 is 2000 r.p.m., required the horsepower if the speed is increased to 4000 r.p m.

7. The demands on a fan running 2000 r.p.m have increased, and it is estimated the fan will deliver the required volume of air if speed is increased to 4000 r.p.m Show why it will be much more economical to replace blower with one designed to deliver the required volume under the original pressure requirements.

8. Required the capacity of an induced fan suitable for the conditions stated in Problem 3, Chapter VIII.

9. Required the power necessary to operate the fan in Problem 7, if its mechanical efficiency is 60 per cent under the specified operating conditions.

CHAPTER X

RECIPROCATING STEAM ENGINES

162. Introductory. — The type of prime mover best suited for a given installation is the one that delivers the required power at the lowest cost, taking into consideration all charges, fixed and operating. These include not only the cost of fuel, labor, supplies, and repairs, but all overhead charges such as interest on the investment, depreciation, maintenance, and taxes. Steam conditions as regards pressure and temperature, size of units, kind of service, nature of the station load curve, and disposition of the exhaust are important factors which must also be considered. Space requirements and continuity of operation are often of vital importance, and may greatly influence the selection of type of prime mover and auxiliary apparatus. In some situations, the gas engine and producer are productive of the highest commercial economy; in others, the choice lies between the reciprocating steam engine and the turbine; frequently the hydro-electric plant offers the best returns; but each proposed installation is a problem in itself, and general rules are without purpose.

The reciprocating steam engine is the most widely distributed prime mover in the power world, and although its field of usefulness has been greatly encroached upon in recent years by the steam turbine and internal combustion engine, it is still an important heat engine and will probably continue to be a factor for years to come. In a general sense, the piston engine is superior to the turbine for variable speed, slow rotative speeds, and heavy starting torque, while the turbine has superseded the engine for large central station units and for auxiliaries requiring high rotative speed. The high-speed turbine in connection with efficient reduction gearing has some advantages over the piston engine for low-speed drives and is to a certain extent replacing the latter in this connection. From a purely thermal standpoint, the Diesel type of internal combustion engine is superior to the steam engine and the turbine is more economical in space requirements, but taking into consideration all of the items affecting the production of power, the reciprocating engine may still prove to be the better investment in many situations.

Improvement in the heat efficiency of the piston engine within the past few years has been remarkable, and single-cylinder units of the

uniflow or **unaflow** type are being operated with steam consumptions lower than that obtained from the older counterflow types of compound units. A 150-hp. Schmidt high-pressure engine with interstage superheating, recently tested in Germany, gave an overall thermal efficiency of 31 per cent (indicated horsepower basis). (See Table 61, paragraph 187.) A few years ago the piston engine appeared to be doomed to the scrap heap, but the unusual economies effected in the later designs have made it once more a formidable competitor of the steam turbine, at least for moderate power requirements and non-condensing service. The recent installation of a 25,000 hp. counterflow piston engine in an English rolling mill is evidence that this type of prime mover is not necessarily limited to small sizes.[1]

Because of the limited space available, and considering the fact that this phase of the subject has been thoroughly covered in text books, no attempt will be made to describe the various types of counterflow piston engines or to analyze the constructive details. The discussion has been limited to the possibilities of the perfect engine and the various factors which affect the performance of the actual mechanism, with a view of selecting the characteristics best suited for a given kind of service.

163. The Ideal or Perfect Engine. — In every heat engine the working fluid goes through a circuit, or cycle, of operation. Beginning at a particular condition, it passes through a series of successive states of pressure, volume, and temperature and returns to the initial condition. An ideally perfect engine, which effects the highest possible conversion of heat into mechanical work for a given cycle, is taken as a standard of comparison for the performance of the actual engine. There are several cycles which approach more or less the action of steam in the actual engine, but the Rankine cycle meets the conditions of most piston engines and for that reason has been adopted as a standard. The various cycles are treated at length in Chapter XXIII and need not be considered here.

In order to realize the ideal Rankine cycle, the walls of the cylinder and the piston must be non-conducting, expansion after cut-off must be adiabatic and carried down to the existing back pressure, the action of the valves must be instantaneous, and steam passages must be sufficiently large to prevent wire drawing. Practically none of these conditions is fulfilled by the actual engine. The various losses which prevent the actual engine from obtaining the efficiency of the ideal are outlined in paragraphs 171 to 181.

The heat supplied, heat consumption, efficiency, and water rate of a perfect engine operating in the Rankine cycle are treated at length in Chapter XXIII and may be summed up as follows:

[1] For a description of this engine see *Power*, Sept. 26, 1922, p. 491.

$$\text{Heat supplied} = H_i - q_n \tag{125}$$
$$\text{Heat converted into work} = H_i - H_n \tag{126}$$
$$\text{Efficiency, } E_r = (H_i - H_n) \div (H_i - q_n) \tag{127}$$
$$\text{Water rate, } W_r = 2547 \div (H_i - H_n) \tag{128}$$

in which

E_r = efficiency on the Rankine cycle with complete expansion,
W_r = water rate on this cycle, lb. per hp-hr.,
H_i = initial heat content of the steam, B.t.u. per lb.,
H_n = final heat content after adiabatic expansion from initial condition to final condition n, B.t.u. per lb.,
q_n = heat of the liquid corresponding to exhaust temperature, B.t.u. per lb.

The average engine seldom expands to the existing back pressure, in which case the work done per lb. of steam supplied is less than if the expansion were complete. The various theoretical quantities for this condition of incomplete expansion (see paragraph 397) may be calculated as follows:

$$\text{Heat supplied} = H_i - q_n, \text{ B.t.u. per lb.} \tag{129}$$

It will be noted that this is the same as for complete expansion.

$$\text{Heat absorbed} = H_i - H_c + v_c (P_c - P_2)/778 \tag{130}$$
$$\text{Efficiency, } E'_r = [H_i - H_c + v_c (P_c - P_2)/778] \div (H_i - q_n) \tag{131}$$
$$\text{Water rate, } W'_r = 2547 \div [H_i - H_c + v_c (P_c - P_2)/778] \tag{132}$$

in which

E'_r = efficiency of the Rankine cycle with incomplete expansion,
H_c = heat content at release pressure P_c after adiabatic expansion, B.t.u. per lb.,
P_c = release pressure, lb. per sq. ft.,
P_2 = back pressure, lb. per sq. ft.,
v_c = specific volume of the fluid under release conditions, cu. ft. per lb. (Other notations as for complete expansion.)
W'_r = water rate of this cycle, lb. per hp-hr.

Direct-acting steam pumps and engines taking steam full stroke have the following theoretical possibilities (see paragraph 399):

$$\text{Heat supplied} = H_i - q_n, \tag{133}$$
$$\text{Heat absorbed} = v_1 (P_1 - P_2)/788 \text{ B.t.u.} \tag{134}$$
$$\text{Efficiency, } E_r'' = v_1 (P_1 - P_2) \div 778 (H_i - q_n) \tag{135}$$
$$\text{Water rate, } W_r'' = 2547 \times 778 \div v_1 (P_1 - P_2) \tag{136}$$

in which

v_1 = specific volume of the steam at pressure P_1, cu. ft. per lb.
P_1 = initial pressure, lb. per sq. ft.,
P_2 = back pressure, lb. per sq. ft.
E_r'' = efficiency of the non-expansion basis,
W_r'' = water rate of this cycle, lb. per hp-hr.
(Other notations as for complete expansion.)

Cylinder Efficiency of an Engine Reveals Avoidable Losses: Power, May 1, 1923, p. 678.

TABLE 56

THEORETICAL EFFICIENCIES AND WATER RATES OF PERFECT ENGINES OPERATING IN THE CARNOT AND RANKINE CYCLES

(Saturated Steam)

Initial Pressure Lb Abs.	Condensing, Back Pressure, 1 In. Hg				Non-condensing, Back Pressure, 14 7 Lb Abs.			
	Efficiency Per Cent		Water Rate Lb per I Hp-hr		Efficiency Per Cent		Water Rate Lb per I.Hp-hr.	
	C	R	C	R	C	R	C	R
50	27.18	24.98	10.13	8.98	9.32	8.98	29.56	28.51
100	31.51	28.47	9.10	7.85	14.70	13.88	19.48	18.22
150	34.10	30.60	8.65	7.26	17.90	16.65	16.46	15.08
200	35.91	31.88	8.41	6.94	20.19	18.60	14.94	13.44
250	37.34	32.93	8.25	6.70	21.97	20.05	14.02	12.42
300	38.51	33.76	8.14	6.52	23.42	21.22	13.39	11.71
400	40.37	35.10	8.04	6.25	25.74	23.07	12.53	10.73
500	41.79	36.06	8.07	6.07	27.54	24.46	12.22	10.10
600	43.00	36.84	8.10	5.94	29.02	25.57	11.98	9.66
800	44.80	38.00	8.24	5.79	31.20	27.20	11.81	9.12
1000	46.30	38.90	8.40	5.67	33.10	28.50	11.80	8.71
1200	47.60	40.00	8.68	5.53	34.50	29.70	11.92	8.40

164. Efficiency Standards. — In order to place the performance of reciprocating engines on a comparable basis and to avoid confusion in the meaning of the terms used in expressing such performance, all tests should be conducted in accordance with Test Code on Reciprocating Engines, as recommended by Power Test Committee of the American Society of Mechanical Engineers. Directions regarding the application, use, and calibration of the various instruments and apparatus used for conducting engine tests, statements as to their accuracy, methods of conducting tests, and definitions of the different terms used in expressing the performance are detailed in the code. The performance of reciprocating engines is stated as follows:

1. Water rate, lb. of steam per unit output per hr.
2. Heat supplied, B.t.u. per unit output per hr.
3. Thermal efficiency, per cent.
4. Mechanical efficiency, per cent.
5. Rankine cycle efficiency, per cent.

The indicator offers the simplest means of measuring the output of a piston engine, and for this reason the performance is usually stated in terms of indicated horsepower. The indicated horsepower is always greater than the net available power by an amount equivalent to the friction of the mechanism. The power actually developed, or brake horsepower, is not readily obtained except for small sizes, and it is customary to approximate this value by deducting the indicated horsepower when running idle from the indicated horsepower when running under the given load. This does not give the true effective power, but is sufficiently accurate for most commercial purposes. (See paragraph 178.) The output of steam turbines and piston engines driving electrical machinery is conveniently stated in electrical horsepower or kilowatts, since the electrical measurements are readily made. The electrical output as measured at the generator terminal gives the net effective work, and automatically deducts the machine losses.

165. Water Rate. — The most generally used measure of the performance of a steam engine is the **water rate**, or lb. of steam supplied at actual condition per unit of output, no correction being made for quality or superheat. This includes condensation from jackets, receivers, and reheater coils, if the engine is of the jacketed or compound type. Since the indicator offers the simplest means of measuring the output, the performance is usually stated in terms of indicated horsepower. Except with small engines where the developed power can be determined by a brake test, the water rate per br.hp-hr. is ordinarily approximated by multiplying the indicated water rate by an assumed mechanical efficiency. The water rate per electrical hp-hr. or per kw-hr. for generator driving sets is readily calculated from the electrical output. By plotting the *total weight* of steam passing through the engine as ordinates, and the output, whether i.hp., br.hp. or kw., as abscissas, the resulting curve is a straight line, or nearly so, and is known as the **Willans line.** If the control is by throttling, the curve is a straight line, and if the engine is of the automatic cut-off type the curve is convex to the X-axis. If the curve is a straight line, the water rate at any load may be calculated by knowing the value of two points on the curve, or of one point and the slope. The Willans line, whether straight or bent upward, may be closely ap-

proximated by the equation

$$W = A + BP + CP^2,[1] \tag{137}$$

in which

W = lb. of steam per hr.,
P = load, i.hp., br.hp., or kw.,
A = lb. of steam per hr. with engine idling,
B, C = constants determined by experiment,
(C = 0 for a straight line)

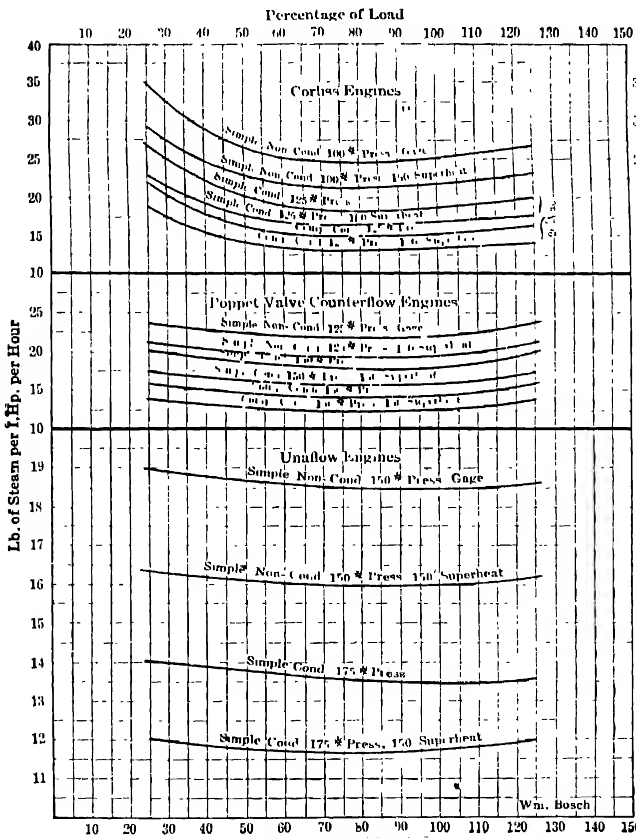


FIG. 247. Typical Water Rates for Various Classes of Engines.

For most engineering purposes, it is sufficiently accurate to assume a straight line for all classes of piston engines. The unit water-rate curve formed by plotting water rates as ordinates against unit output is of great

[1] Engineering Thermodynamics, Lucke, p. 386.

importance in visualizing the performance of the engine at various loads. The aim of the engine builder is to have this unit curve as flat as possible, so that the efficiency may be the same at all loads within the working range. A number of typical water-rate curves will be found in this chapter under various headings. If the initial pressure, quality, and back pressure were constant for all conditions of operation, the water rate would be a true measure of the heat efficiency, but since this is not the case the water rate under actual conditions is of little value in comparing performances. The water rate may be used as a means of comparison, provided suitable corrections are made for pressures and quality, but this procedure is not common. Where it is desired to correct to any assumed standard conditions, as in contract guarantee or acceptance tests, the corrections applied should be agreed upon by the interested parties because there are no standard correction factors applicable to all makes and sizes of engines.

166. Heat Supplied. — Heat supplied for engines and turbines is expressed as the total heat content of the steam supplied less the heat of the liquid at exhaust pressure. The heat of the liquid at exhaust is sometimes known as the **ideal feedwater temperature.** The temperature of the liquid for a non-condensing engine exhausting at standard atmospheric pressure is 212 deg. fahr., and that of a condensing engine exhausting against an absolute back pressure of 2 lb. is 126 deg. fahr. and so on. The heat supplied, referred to any unit of output, is defined as the heat input per unit of output, for example: B.t.u. per i.hp-hr.; B.t.u. per kw-hr., etc., and is a true measure of the heat economy. The heat of the liquid at exhaust pressure is dependent upon pressure only, and therefore is independent of any heat which may be collected in the condensation from jackets or reheater-receiver coils. Utilization of the heat in this condensate increases the overall station economy but is not considered in measuring the net heat supplied to the engines.

Example 31. -- (1) A compound condensing engine develops a brake hp-hr. on a steam consumption of 8.5 lb. initial pressure 200 lb. abs., superheat 250 deg. fahr., exhaust pressure 0.5 lb. abs., release pressure 2 lb. abs. (2) The same engine when using wet steam develops a brake hp-hr. on a steam consumption of 12 lb. per hr., initial pressure 150 lb. abs., quality 98 per cent, exhaust pressure 2 lb. abs., release pressure 4 lb. abs. Compare the performance of the two engines on a " heat supplied " basis.

Solution. — The heat content of the superheated steam and that of the liquid is found from steam tables to be $H_i = 1332$; $q_n = 48$.

The heat content of the wet steam may be taken directly from the Mollier diagram or calculated

$$H_i = 0.98 \times 863.2 + 330.2 = 1176.1; \; q_n = 94.$$

Heat supplied to superheated steam engine

$$8.5\ (1332 - 48) = 10{,}914 \text{ B.t.u. per br.hp-hr.}$$

Heat supplied to saturated steam engine =

$$12\ (1176.1 - 94) = 12{,}985 \text{ B.t.u. per br.hp-hr.}$$

Economy of superheated- over saturated-steam engine

(1) in water rate: $100\ (12 - 8.5) \div 12 = 29.2$ per cent.
(2) in heat supplied: $100\ (12{,}985 - 10{,}914) \div 12{,}985 = 15.9$ per cent.

167. Thermal Efficiency. — The thermal efficiency of a steam engine is the ratio of the heat equivalent of the work done to that supplied, measured above the heat of the liquid at exhaust pressure. It may be expressed as indicated, brake, or combined thermal efficiency, depending upon whether the work done is based upon the indicated load, brake load or combined output of engine and generator, respectively. Since the heat equivalent of one hp. is 2547 B.t.u. per hr., the indicated, or brake thermal efficiency E_t may be expressed:

$$E_t = 2547 \div W(H_i - q_n) \tag{138}$$

in which

$$W = \text{lb. steam per i.hp-hr. or br.hp-hr.}$$

H_i and q_n as in equation (125)

Since the heat equivalent of one kw. is 3415 B.t.u. per hr., the combined thermal efficiency of engine and generator E_t is

$$E_t = 3415 \div W_1\ (H_i - q_n) \tag{139}$$

in which

$$W_1 = \text{lb. steam per kw-hr.}$$

Other notations as in equation (138)

Example 32. — Calculate the thermal efficiencies of the two engines using the data in the preceding example.

Solution. — Superheated steam engine.
$E_t = 2547 \div 8.5\ (1332 - 48) = 0.233 = 23.3$ per cent.
Saturated steam engine.
$E_t = 2547 \div 12\ (1176.1 - 94) = 0.196 = 19.6$ per cent.

The thermal efficiency of the actual engine varies from 5 per cent in the poorest grade of non-condensing single-cylinder single-valve design operating with saturated steam, to 31 per cent in the highest grade multi-expansion engine with high-pressure highly-superheated steam, the best

recorded performance to date. As far as thermal efficiency is concerned, the non-condensing piston engine still leads the non-condensing turbine for sizes under 2000 hp.

168. Mechanical Efficiency. — The ratio of the brake horsepower to the indicated power is the mechanical efficiency of the engine; the ratio of the electrical horsepower to the indicated power is the mechanical efficiency of the engine and generator combined; and the ratio of the pump horsepower to the indicated power of the engine is the mechanical efficiency of the engine and pump combined. The percentage of work lost in friction is, therefore, the difference between 100 per cent and the mechanical efficiency in per cent. (See also paragraph 178.)

The mechanical efficiency of piston engines at rated load varies from 85 per cent for the cheaper grades of engines to 95 per cent and even 98 per cent for the better types. With highly superheated steam, the mechanical efficiency is apt to be lower than with saturated steam unless careful consideration has been paid to the system of lubrication. The friction horsepower decreases somewhat with the increase in size of engine and is considerably higher with newly installed engines than after they have been in service a year or so. (See *Power*, Oct. 25, 1921, p. 652.) Generator efficiencies at full load vary from 86 per cent for the 15-kw. size to 94 per cent for units of 2000 kw. rated capacity. The overall or combined efficiency at rated load varies from 75 per cent for small units to 93 per cent for larger ones. The generator efficiency of very large turbo-alternators, 25,000 kw. rated capacity or more, is in the neighborhood of 97 per cent. The efficiency at fractional loads for a specific case is illustrated in Fig. 248.

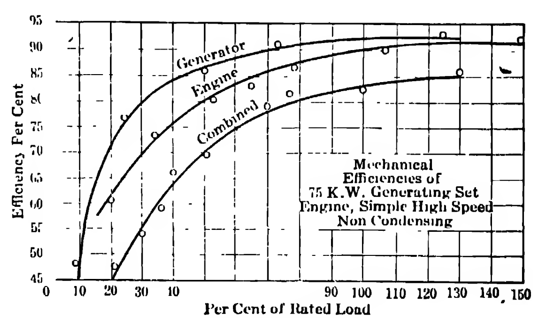


FIG. 248.

Lucke, *Engineering Thermodynamics*, p. 370, states that the mechanical efficiency of the piston engine is independent of the speed and that it may be expressed

$$E_m = 1 - K_1 - K_2/m \tag{140}$$

in which

K_1 = constant, varying from 0.02 to 0.05,
K_2 = constant, varying from 1.3 to 2.0,
m = mean effective pressure, lb. per sq. in.

169. Rankine Cycle Efficiency. — The degree of perfection of an engine, or the extent to which the theoretical possibilities are realized, is the ratio of the thermal efficiency of the actual engine to that of an ideally perfect engine working in the Rankine cycle with complete expansion. This is called the Rankine efficiency, or Rankine cycle ratio, and is expressed on the i.hp., br.hp. or kw. basis. It is the accepted standard for comparing the performance of steam engines.

If E = Rankine efficiency[1]
E_t = thermal efficiency of the actual engine
E_r = efficiency of the ideal engine working in the Rankine cycle with complete expansion.

Then $$E = E_t/E_r \tag{141}$$

From equation (138)

$$E_t = 2547 \div W(H_i - q_n)$$

And from equation (128)

$$E_r = (H_i - H_n) \div (H_i - q_n)$$

Whence

$$E = 2547/W(H_i - q_n) \div (H_i - H_n)/(H_i - q_n) \tag{142}$$
$$= 2547 \div W(H_i - H_n) \tag{143}$$

The Rankine cycle efficiency may also be expressed as the ratio of the actual unit water rate to that of the perfect engine working through the same pressure and temperature range.

Example 33. — Determine the Rankine cycle efficiency of the two engines specified in Example 31.

Solution. — Superheated steam engine.
$E = 2547 \div 8.5\ (1332 - 908) = 0.706 = 70.6$ per cent (br.hp. basis)
Saturated steam engine.
$E = 2547 \div 12\ (1176 - 898) = 0.763 = 76.3$ per cent (br.hp. basis)

Rankine cycle efficiencies (i.hp. basis) for various types of engines under average steam conditions and at rated load range are approximately as shown in Table 57.

[1] The term "efficiency" without qualification when applied to the performance of engines is usually considered to mean Rankine cycle efficiency.

TABLE 57

RANKINE CYCLE EFFICIENCIES, PER CENT

	Saturated Steam		Superheated Steam	
	Non-condensing	Condensing	Non-condensing	Condensing
* Simple, single valve	50–65	38–45	65–75	50–65
* Simple, multi-valve	65–75	42–63	72–82	60–75
Simple, uniflow	70–80	65–70	75–85	70–80
* Compound	70–80	63–73	75–85	68–80
* Triple expansion	72–82	70–76	75–86	70–82

* Counterflow

The piston engine seldom expands the steam down to the existing back pressure, but releases from 2 to 5 lb. above this point in condensing engines and from 15 to 20 lb. above in non-condensing engines. The ideal cycle corresponding to this condition is the Rankine cycle with incomplete expansion. The ratio of the thermal efficiency of the actual engine to that of the ideal engine working in the incomplete cycle is a true measure of the degree of perfection of the engine under the given conditions. This rate is sometimes called *cylinder efficiency* and may be expressed as

$$E' = \frac{2547}{W\left[(H_1 - H_c) + (P_c - P_2)x_c u_c \div 778\right]} \tag{144}$$

See equations (138) and (132).

Example 34. — Determine the cylinder efficiency of the two engines specified in Example 31.

Solution. — Superheated steam engine:

$$E' = \frac{2547}{8.5\left[1332 - 980 + \frac{144}{778}(2.0 - 0.5)0.866 \times 173.5\right]}$$
$$= 0.761 = 76 \text{ per cent.}$$

Saturated steam engine:

$$E' = \frac{2547}{12\left[1176 - 935 + \frac{144}{778}(4 - 2)0.808 \times 90.5\right]}$$
$$= 0.808 = 80.8 \text{ per cent.}$$

Summing up the various efficiencies for the two cases analyzed in paragraphs 166 to 169:

	Saturated Steam Engine	Superheated Steam Engine
Pressure, lb. per sq. in., absolute:		
Initial	150	200
Release	4	2
Condenser	2	0.5
Degree of superheat, deg. fahr.	0.98*	250
Steam consumption, lb. per br hp-hr.:		
Actual engine	12.00	8.50
Ideal engine, Rankine cycle, with incomplete expansion	9.69	6.46
Ideal engine, Rankine cycle, with complete expansion	9.16	6.00
Ideal engine, Carnot cycle	10.59	.
Thermal efficiency, per cent:		
Actual engine	19.6	23.3
Ideal engine, Rankine cycle, with incomplete expansion	24.3	30.7
Ideal engine, Rankine cycle, with complete expansion	25.8	33.3
Ideal engine, Carnot cycle	28.3	..
Heat consumption, B.t.u. per br hp-hr.:		
Actual engine	12,985	10,914
Ideal engine, Rankine cycle, with incomplete expansion	190.4	154.6
Ideal engine, Rankine cycle, with complete expansion	174.8	138.4
Ideal engine, Carnot cycle	152.5	.
Rankine efficiency, per cent	76.3	70.6
Cylinder efficiency, per cent	80.8	76.1

* Quality

170. Commercial Efficiencies. — There is no accepted standard for rating the commercial efficiency of an engine or turbine. The various measures used in this connection, such as **B.t.u. of fuel fired per hp- or kw-hr., lb. of standard coal per br.hp-hr., cents per hp. per year,** and the like, include the economy of the boiler and auxiliaries and are not a true indication of the performance of the engine alone. From a commercial standpoint, it is important to know the weight of coal required to develop a hp-hr., taking into consideration all of the losses of transmission and conversion, and a knowledge of the **overall efficiency** from switchboard to coal pile is of value in basing the cost of power; but these items are in reality measures of the **plant economy** and are of little value in comparing the performance of the prime mover.

Commercial Efficiency in Transformation and Distribution of Energy: C. E. Lucke, Mech. Engrg., June, 1924, p. 317.

171. Heat Losses in the Steam Engine. — The principal losses which tend to lower the efficiency of the steam engine and which prevent it from realizing the performance of the ideal engine are due to:

(*a*) Cylinder condensation;
(*b*) Leakage;
(*c*) Clearance volume;

(d) Incomplete expansion;
(e) Wire drawing;
(f) Friction of the mechanism;
(g) Presence of moisture in the steam at admission;
(h) Radiation, convection, and minor losses.

172. Cylinder Condensation. — The weight of steam apparently used per revolution, as determined from the indicator card, or the **indicated steam consumption**[1] (see paragraph 403) is considerably less than that actually supplied. The difference or **missing quantity** is due chiefly to **cylinder condensation.** This is by far the greatest loss in the steam engine with the exception of that inherent in the ideal engine. When steam is admitted to the cylinder of a counterflow engine, a considerable portion of the heat is given up to the comparatively cool skin surface of the cylinder walls. If the steam is saturated at admission, this heat absorption causes condensation, or *initial condensation* as it is called; if superheated at admission, the temperature is lowered to a corresponding point. After cut-off, heat continues to be given up to the walls until the temperature of the steam falls below that of the skin surface, when the process is reversed and part of the heat is returned to the steam. With saturated steam the heat absorption causes **condensation during expansion,** and, according to the accepted theory, the heat rejected causes **re-evaporation during expansion.** According to the investigation of M. H. Barker, engineer of the Hooven, Owens, Rentchler Co., the time from cut-off to release is too short for the water of condensation to absorb the heat from the cylinder walls, and the departure of the expansion line from adiabatic is due to leakage. (See *Power*, Oct. 9, 1923, p. 567.) With superheated steam an equivalent heat exchange takes place, though to a less marked degree. Unless the cylinder is of a compound series, the heat absorbed from the cylinder walls during exhaust does no useful work and is lost. With counterflow engines, the exhaust steam must be returned in order to pass through the exhaust valves located at the ends of the cylinder, thereby cooling the clearance surfaces, which in turn condense a part of the incoming steam on the next working stroke. In the uniflow engine (see paragraphs 190 and 197) the expanded and cooled steam does not flow over the clearance surfaces but is discharged through ports in the center of the cylinder. Furthermore, during compression, the exhaust steam trapped in the cylinder is compressed against a jacketed head, and that portion remaining in the clearance space is heated by compression to a temperature practically that of the initial steam. When the admission valve opens again to start the next stroke, live steam rushes

[1] Also called the *steam accounted for by the diagram or diagram steam.*

in and encounters the steam which is compressed to a high temperature, and practically no initial condensation takes place. Cylinder condensation and leakage, measured as the proportion of the mixture present, varies with the type and size of the engine, length of cut-off, valve design, temperature range, location of ports and port passages, jacketing, lagging, and other variables. It ranges from 18 to 60 per cent in the counterflow engine, and from 12 to 25 per cent in the uniflow. In the uniflow engine, however, the curve is much flatter, as will be seen from an inspection of the curves in Fig. 249. These curves are based on indicator cards taken from a number of non-condensing engines and are, therefore, applicable only to the particular types and sizes tested and for the conditions under which the tests were made, but are of interest in showing the influence of ratio of expansion on cylinder condensation and leakage losses for the two types of engines.

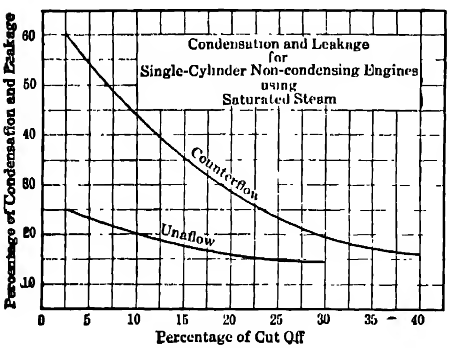


FIG. 249.

The empirical formulas which may be used for calculating the extent of these losses, and which involve the various influencing factors, are unwieldy and only approximately accurate. One of the most satisfactory formulas of this class is that deduced by R. C. H. Heck, "The Steam Engine and Turbine," 1911, p. 175.

The various heat exchanges between the working fluid and the cylinder walls, including cylinder condensation and leakage, are approximately determined by transferring the indicator diagram to the temperature entropy chart. (See paragraph 401.) For use and application of the temperature-entropy diagram in engine tests, consult *Power*, Dec., 1907, p. 834; Jan. 21, 1908, p. 96; Jan. 28, 1908, p. 145.

A comparatively simple method for approximating cylinder condensation and leakage losses is given by J. Paul Clayton, *Bulletin No.* 58, 1912, *University of Illinois*, and consists in transferring the indicator diagram to logarithmic cross-section paper. By means of the logarithmic diagram Clayton found that, (1) free from certain abnormal influences, expansion and compression take place in the cylinder substantially according to the law $PV^n = C$; (2) the value n bears a definite relation in any given cylinder to the proportion of the total weight of steam mixture which was present as steam at cut-off; (3) the relation of the value n to the value x_c (quality of steam at cut-off) for the same class of cylinder, as

regards jacketing, is practically independent of engine speed and of cylinder size; and (4) by means of the experimentally determined relation of x_n and n, the actual steam consumption may be obtained from the indicator card to well within 4 per cent of the true value. The curves in Fig. 701 were plotted on logarithmic cross-section paper from the pressure-volume diagrams of a 12-in. by 24-in. Corliss engine, and illustrate Mr. Clayton's method of analysis. The curves in Fig. 702 show the relation between quality x_c and exponent n for a given set of conditions. See also paragraph 400.

173. Leakage of Steam. — The loss due to leakage is a variable factor depending upon the design and condition of the engine, and is greater with saturated than with superheated steam. According to the investigation of M. L. Barker, about one out of three engines has tight valves. (See *Power*, Oct. 30, 1923, p. 693.) The usual method of measuring leakage past the valves and piston while the engine is at rest is likely to give erroneous results, as demonstrated by Callender and Nicolson (Peabody, "Thermodynamics," p. 351) in tests made on a high-speed automatic balanced-valve engine and on a quadruple expansion engine with plain unbalanced slide valves. With the engines at rest, they found that the leakage past valves and piston was insignificant, but, when in operation, the leakage from the steam chest into the exhaust was considerable. It was thought that a large proportion of the leakage was probably in the form of water formed by condensation of steam on the seat uncovered by the valve.

According to the report of the Steam Engine Research Committee (*Engrg.* Lond., March 24, 1905, p. 393), leakage through a plain slide valve is independent of the speed of the sliding surfaces, and directly proportional to the difference in pressure on the two sides; with well-fitted valves the leakage is never less than 4 per cent of the volume of steam entering the cylinders, and is often greater than 20 per cent.

The various leakage losses may be approximated by transferring the indicator diagram to logarithmic cross-section paper. Figure 703 shows the application of the logarithmic diagram to a specific case and illustrates this method of determining leakage losses. See paragraph 400.

Leakage Past Piston Valves: Engrg., Feb. 9, 1912.

174. Clearance Volume, and Compression. — The portion of the cylinder volume which is not swept through by the piston but which is nevertheless filled with steam when admission occurs is called the **clearance volume.** It is the space between the end of the piston when on dead center and the inside of the valves covering the ports. In counterflow engines it varies from about 2 per cent of the piston displacement

in very large engines with short steam passages, to 10 per cent or more in small high-speed engines. In uniflow engines it ranges from 1/2 to 5 per cent, the lower value for the largest engines with single-beat poppet valves.

The extent of surface in the clearance space greatly influences the amount of cylinder condensation, since the piston is moving slowly near the end of the cylinder, and the time of exposure of the steam to these surfaces is comparatively long. The greater part of the cylinder condensation usually occurs in the clearance space; therefore the steam passages and clearance space should be designed so as to present a minimum amount of surface consistent with the proper cushioning volume for smooth operation. Theoretically, if steam is compressed adiabatically to the initial pressure there is no loss due to clearance, but in counterflow engine practice compression carried to initial pressure does not necessarily improve the economy. For a constant time element, the shorter the cut-off the greater will be the ratio of the weight of cushion steam to that of the steam supplied and hence the greater the loss. In large slow-moving engines the loss due to clearance may be greater than that in high-speed, short-stroke engines because of the longer time of exposure to the clearance surface. According to Professor J. Stumpf (The Unaflow Engine, 1922, p. 42), " (1) for the same initial and back pressure, m.e.p., and best compression, the theoretical steam consumption increases almost linearly with the clearance volume. (2) The volume loss increases with increasing initial pressure, decreases with increasing back pressure and m.e.p., and becomes a minimum for a certain length of compression. (3) The clearance volume loss is zero if expansion reaches the back pressure and compression rises to the initial pressure. (4) For a given initial pressure, back pressure, m.e.p., and length of compression, the clearance volume must be such that the change of pressure during expansion is equal to the change of pressure during compression."

The ratio of expansion is a function of the clearance volume; for example, an engine cutting off at one fifth, neglecting clearance, has an apparent ratio of expansion of 5, but if the clearance volume is 10 per cent the *actual* ratio is only 3.66.

In high-speed engines a certain amount of compression is desirable for its cushioning effect irrespective of its influence on economy. According to Professor J. Stumpf, " For a given initial pressure, mean effective pressure, and clearance volume, the lowest steam consumption is obtained if the length of compression is made 100 per cent and the back pressure is chosen so as to make the change of total heat during expansion equal to the change of total heat during compression."

A series of tests by Professor Jacobus (*Trans. A.S.M.E.*, 15–918) on a

10-in. by 11-in. high-speed automatic engine at Stevens Institute show decreasing economy with increase of compression, the initial pressure, cut-off, and release remaining constant. The results were as follows:

Proportion of initial pressure up to which the steam is compressed	$\frac{3}{8}$	$\frac{2}{3}$	Full
Steam, lb. per i.hp-hr..	34.8	36.7	38

Tests by Carpenter (*Trans. A.S.M.E.*, 16–957) on the high-pressure cylinders of the Corliss engine at Sibley College gave:

Compression, per cent.	11.4	25	35.2
Brake hp.	30	29	26
Steam, lb. per br.hp-hr.	33	33.3	34

Tests made by A. H. Klemperer on a 7.1-in. by 17.7-in. Corliss engine, at Dresden, gave decreasing steam consumption for increase in compression volume up to about twice that of the clearance beyond which the water rate increased with the increase in compression. (*Zeit. d. Ver. deut. Ingr.*, Vol. 1, 1905, p. 797.)

Tests made by Professor Boulvin on a 9.8-in. by 19.7-in. Corliss engine at the University of Ghent gave results agreeing with those of Klemperer. (*Revue de Mecanique*, 1907, Vol. XX, p. 109.)

175. Loss Due to Incomplete Expansion. — In the perfect or ideal engine, maximum economy is effected by expanding the steam down to the existing back pressure. The increase in mean effective pressure, however, resulting from complete expansion, particularly for low back pressure and high initial pressure, is comparatively small and necessitates the use of extremely large cylinders for a given power output. In the actual engine, the m.e.p. is less than in the perfect engine for the same conditions; therefore, in order to obtain more power for reasonable cylinder dimension, expansion is not carried to the back-pressure line but to some point above this limit. Furthermore, in the actual engine, the greater the ratio of expansion the greater will be the loss due to cylinder condensation, and at some point in the expansion the loss just balances the theoretical gain and any further expansion beyond this point will be at the expense of economy. This critical point varies with the design of engine, initial pressure, and quality of steam, and the back pressure. In the ordinary single-cylinder non-condensing counterflow engine using saturated steam, it corresponds to approximately one-quarter cut-off. As will be shown later, the ratio of the expansion may be increased with reduced condensation losses, or the equivalent, by compounding, superheating, or employing the uniflow principle.

176. Loss Due to Back Pressure. — In the perfect engine, for a given initial pressure and quality of steam and a fixed ratio of expansion or m.e.p., any reduction in back pressure will result in increased horsepower directly proportional to the m.e.p. Conversely, any increase in back pressure will result in correspondingly decreased horsepower. Thus it will be seen that more power can be realized for a given weight of steam by this reduction of back pressure, or, for a given power, less steam need be furnished the engine. In the actual engine this law holds true within certain limits only. For example, a single-cylinder engine designed for non-condensing service will show decreased economy almost directly proportional to the increase in back pressure for pressures above atmospheric, but when the back pressure is reduced by condensing, the gain in economy is less as the degree of vacuum increases. This is due partly to leakage at the higher vacua and to increased cylinder condensation because of the increased temperature range within the cylinder. This is true for all classes and types of piston engines, but the degree of departure from the performance of the ideal engine is more marked in the single-cylinder counterflow than in the compound counterflow or single-cylinder uniflow engine. Professor J. Stumpf, in "The Unaflow Steam Engine," 1922 Edition, p. 43, proves deductively that in every piston engine there is a back pressure for each set of operating conditions under which maximum economy can be realized. This **critical back pressure** is a function of the clearance volume, initial pressure, load, and length of compression.

Professor Stumpf has experimented with exhaust nozzles with the view of making use of the kinetic energy of the exhaust to reduce the cylinder back pressure. This has proved successful on the Prussian State Railways in connection with multi-cylinder uniflow engines. Considerable experimental work of this nature is under way in America, but the art has not been developed to the point where stationary units are employing this principle. For a mathematical analysis of the exhaust ejector consult "Using Exhaust Energy in Reciprocating Engines," by J. Stumpf, *Mech. Engrg.*, June, 1922, p. 369.

177. Loss Due to Wire Drawing. — Wire drawing, or the drop in pressure due to the resistances of the ports and passages, has the effect of reducing the output and the economy of the engine to some extent, since the pressure within the cylinder is less than that at the throttle during admission and greater than discharge pressure at exhaust. The steam may be dried to a small extent during admission, but because of the drop in pressure the **heat availability** is reduced. The loss in available heat may be calculated as shown in paragraph 391. In single-valve engines the effects of wire drawing are decidedly marked and the true points of cut-off and release are sometimes difficult to locate on the indi-

cator card. In engines of the Corliss, poppet, or gridiron-valve type, the effects are hardly noticeable.

178. Loss Due to Friction of the Mechanism. — The difference between the indicated horsepower and that actually developed is the power consumed in overcoming friction, and varies from 4 to 20 per cent of the indicated power, depending upon the type and condition of the engine and the method of lubrication. Engine friction may be divided into (1) initial or no-load friction and (2) load friction. The stuffing-box and piston-ring friction is practically independent of the load, while that of the guides, bearings, and the like increases with the load. The curves in Fig. 250 show the relation between friction horsepower and developed horsepower for a few types of engines. The distribution of the frictional losses in a number of engines is given in Table 58. See also paragraph 168.

TABLE 58

DISTRIBUTION OF FRICTION IN SOME DIRECT-ACTING STEAM ENGINES

(Thurston)*

Parts of Engines where Friction is Measured	Percentage of Total Engine Friction				
	"Straight" Line" Balanced Valve	"Straight" Line" Unbalanced Valve	Traction Engine Locomotive Valve Gear	Automatic Balanced Valve	Condensing Engine Balanced Valve
Main bearings .	47.0	35.4	35.0	41.6	46.0
Piston and piston rod	32.9	25.0	21.0		
Crank pin. . .	6.8	5.1	13.0	49.1	21.0
Crosshead and wrist pin.	5.4	4.1			
Valve and valve rod.	2.5	26.1	22.0	9.3	21.0
Eccentric strap. .	5.4	4.0			
Link and eccentric			9.0		
Air pump.					12.0
	100.0	100.0	100.0	100.0	100.0

* "Friction and Lost Work in Machinery," p. 13.

179. Moisture at Admission. — The presence of moisture in the steam pipe is due to condensation caused by radiation or to priming at the boiler. Unless removed by some separating device between boiler and engine, the amount of moisture entering the cylinder may be from 1

to 5 per cent of the total weight of steam, and the work done per lb. of fluid is correspondingly reduced. This loss should not be charged against the engine, however, and its performance should be reckoned on the dry-steam basis. Experiments reported by Professor R. C. Carpenter (*Trans. A.S.M.E.*, 15–438) in which water in varying quantities was introduced into the steam pipe, causing the quality of the steam to range from 99 per cent to 57 per cent, showed that the consumption of *dry steam* per i.hp-hr. was practically constant, the water acting as an inert quantity.

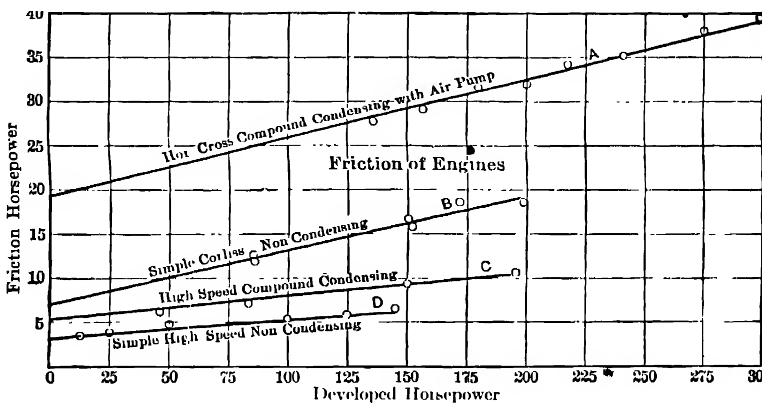


FIG. 250. Typical Curves of Engine Friction.

An efficient separator will remove practically all the entrained water. The presence of large quantities of water in the cylinder is apt to wreck the engine unless it is provided with large automatic snifting valves. Moisture carried from the boiler contains many of the impurities found in the feedwater and is apt to foul the valves and fittings in the pipe line.

180. Radiation and Minor Losses. — The heat loss, commonly called "radiation," from the cylinder steam chest, piston rod, and valve stems, to the surroundings has the effect of increasing the cylinder condensation. In jacket engines this loss may be approximated by the quantity of steam condensed in the jacket when the engine is not running. In unjacketed engines the loss is practically undeterminable since the heat exchange between the cylinder walls and the steam is exceedingly complex. The heat loss due to radiation, measured in terms of the total heat supplied, varies from 0.2 per cent in very large units with efficiently lagged cylinders and steam chests, to approximately 2 per cent in small engines as ordinarily insulated.

181. Heat Lost in the Exhaust. — Most of the heat supplied to the engine is rejected to the exhaust; this varies from 70 per cent in the most economical type of prime mover to 95 per cent in the poorer types.

If the exhaust steam is used for heating or other useful purposes, the heat chargeable to power is the difference between the heat supplied and that utilized from the exhaust, and amounts to approximately 2800 B.t.u. per i.hp-hr. or 4000 B.t.u. per kw-hr. In passing through a prime mover, heat is abstracted from the steam by:

(1) Conversion of part of the heat into mechanical energy,
(2) Loss to the surroundings.

If A = heat converted into work, B.t.u. per lb. of steam,
W, W' and W_1 = water rate, lb. per i.hp-hr., br.hp-hr. and kw-hr., respectively,
e' and e_1 = mechanical efficiency of the engine and combined engine-generator, respectively,

$$\text{Then } A = \frac{2547}{W} = \frac{2547}{e'W'} = \frac{3415}{e_1 W_1} \tag{145}$$

Considering H_1 as the initial heat content, B.t.u. per lb. above 32 deg. fahr., and H_r as the loss due to radiation, B.t.u. per lb., the heat content H_2 per lb. of exhaust will be

$$H_2 = H_1 - H_r - A \tag{146}$$

As previously stated, the heat loss due to radiation in terms of the total heat supplied varies from 0.3 per cent in very large units with efficiently lagged cylinders and steam chests to approximately 2 per cent in small engines of 25 hp. rated capacity. An average value of 1 per cent may be assumed for most practical purposes.

If the exhaust contains moisture, as is usually the case, we have

$$H_2 = x_2 r_2 + q_2, \tag{147}$$

in which

x_2 = quality of the exhaust,
r_2 = latent heat corresponding to exhaust pressure,
q_2 = heat of the liquid at exhaust pressure.

Combining equations (146) and (147) and reducing

$$x_2 = (H_1 - H_r - q_2 - A) \div r_2 \tag{148}$$

If the exhaust is superheated

$$H_2 = r_2 + q_2 + C_m t_2',$$

in which

C_m = mean specific heat of the superheated steam at exhaust pressure,
t_2' = degree of superheat of the exhaust steam, deg. fahr.

The net heat, H_p, chargeable to power is

$$H_p = W(A + H_r), \text{ B.t.u. per i.hp-hr.} \tag{149}$$

All of the heat of the exhaust is not available for commercial heating purposes or process work, because of the condensation losses in the exhaust main. The extent of the latter depends upon the size and length of main, rate of flow, and efficiency of the pipe covering. Representing this loss by H_x(B.t.u. per lb. of steam), and assuming that it is charged against power, the total heat chargeable to power is

$$H_p = W(A + H_r + H_x) \tag{150}$$

and the equivalent water rate, for power only, W_p, is

$$W_p = W(A + H_r + H_x) \div H \tag{151}$$

in which

H = net heat supplied to the engine, B.t.u. per lb.

For output expressed in terms of br.hp. or kw., substitute W' and W_1, respectively, in place of W.

Very little information is available relative to the quality of exhaust as determined by actual test, but such as has been published is in accord with the results calculated from equation (147).

Example 35. — A 23-in. by 16-in. simple engine, direct connected to a 200-kw. generator installed at the Armour Institute of Technology, uses 35 lb. of steam per i.hp-hr. at full load; initial pressure, 115 lb. abs., back pressure, 17 lb. abs., initial quality, 98 per cent.

Calculate the quality of the exhaust, assuming a radiation loss of 1 per cent, and determine the amount of heat chargeable to power.

Solution. — From steam tables

$H_1 = xr + q = 0.98 \times 879.8 + 309 = 1171$; $r_2 = 965.6$; $q_2 = 187.5$
$A = 2547/35 = 72.8$.

By assumption, $H_r = 0.01 \times 1171 = 11.7$
Substituting these values in equation (147) and reducing

$$x_2 = \frac{1171 - 11.7 - 187.5 - 72.8}{965.6} = 0.933 \text{ or } 93.3 \text{ per cent.}$$

(Actual calorimeter tests gave a quality of 92.5 per cent, indicating a somewhat larger radiation loss than the assumed value of 1 per cent.)

Total heat, H_p, chargeable to power (equation 150).

$$H_p = 35\ (72.8 + 11.7 + H_x), \text{ B.t.u. per i.hp-hr.}$$

H_x varies with the size and length of the exhaust main and the efficiency of the covering. Assume that in this particular installation the loss

amounts to the equivalent of 2 per cent of the heat content of the throttle steam. With this assumption we have, as the heat chargeable to power,

$$H_p = 35(72.8 + 11.7 + 23.4) = 3776.5, \text{ B.t.u. per i.hp-hr.}$$

Assuming that the condensation from the heating system, including that exhausted from the engine, is returned to the boiler at a temperature of 192 deg. fahr., the net heat supplied per lb. of steam is

$$H = 1171 - (192 - 32) = 1011 \text{ B.t.u.}$$

And the equivalent water rate for *power only* is

$$3776.5 \div 1011 = 3.73 \text{ lb. per i.hp-hr.}$$

The low fuel consumption for power when the exhaust steam is used for heating purposes is at once apparent.

If steam is extracted at some stage between the high-pressure inlet and the exhaust outlet, as from the receiver of a compound engine, the procedure is the same as previously described except that the water rate up to the point of extraction must be taken instead of the water rate for full expansion. See example 44.

Finding the Cost of Exhaust as Compared to Live Steam. Power, May, 13, 1924, p. 759.

182. Methods of Increasing Economy. — Various methods have been adopted for bettering the economy of piston engines; among them may be mentioned:

(*a*) Increasing initial pressure,
(*b*) Increasing rotative speed,
(*c*) Decreasing back pressure by condensing,
(*d*) Superheating,
(*e*) Use of steam jackets,
(*f*) Reheating receivers,
(*g*) Compounding,
(*h*) Use of uniflow or straight-flow cylinders,
(*i*) Use of binary fluids.

183. Increasing Initial Pressure. — A glance at the curves in Fig. 251, which visualize the relation between thermal efficiency and varying initial pressures for the perfect engine working in the Rankine cycle, shows that with saturated steam the efficiency increases with the initial pressure. Up to a pressure of 250 lb. abs., there is a marked increase in efficiency, but beyond this point the gain is at a gradually decreasing rate. Thus, in raising the pressure from 50 to 250 lb., a range of 200 lb., the efficiency is increased 31.6 per cent for the condensing and 55 per cent for the non-condensing engine, while raising the pressure from 800 to 1200 lb., a range of 400 lb., increases the efficiency of the condensing engine but 5.25 per

cent and that of the non-condensing engine but 9.1 per cent. The performance of the actual engine also shows increased efficiency with higher initial pressures, but to a lesser degree than with the perfect engine. Unless the engine is designed for high pressures there is a point beyond which leakage past pistons and valves and cylinder condensation offset the thermal gain. This point of maximum heat economy varies with the type and construction of the engine and ranges from 165 lb. abs. in the ordinary type of counterflow engine designed for moderate pressures, to 415 lb. abs. in the best type of single-acting uniflow engines. Small

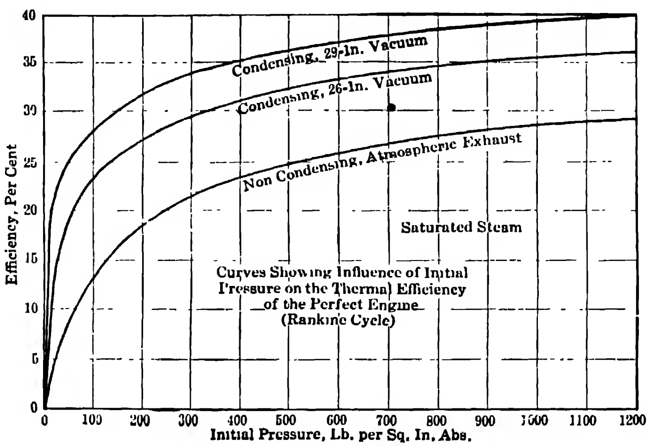


FIG. 251

experimental engines of the compound type designed for high pressures have shown increased efficiency up to the 800 lb. initial pressure, but the gain at pressures above 400 lb. were not commensurate with the increased first cost of the plant equipment. This applies strictly to saturated steam. With high initial superheat and intermediate superheating, the heat economy appears to be limited only by the maximum pressure and temperature that the materials can withstand. In piston-engine practice in this country, the best overall plant efficiency, taking into consideration both fixed and operating costs, is obtained through the use of moderate pressures and superheat. In some steam turbines which are now in course of construction, and which are to operate on a modified regenerative cycle with intermediate reheating, it is proposed to utilize initial pressures of 1200 lb. per sq. in. and a total temperature of 750 deg. fahr. Pressures over 400 lb. per sq. in. are frequently employed in small engines where heat economy is secondary in importance to compactness and high power ratings.

The pressures commonly found in American practice are substantially as follows:

Type of Engine	Range in Pressure (Gage)	Average
Simple slow-speed (standard type)	60–120	90
Simple high-speed (standard type)	70–125	100
Simple, uniflow (non-condensing)	115–225	160
Simple, uniflow (condensing)	125–225	175
Compound high-speed, non-condensing	100–180	150
Compound high-speed, condensing	100–180	150
Compound slow-speed, condensing	125–200	170
Triple expansion, condensing	140–250	200
Quadruple expansion, condensing	175–300	250

In correcting performance data from observed initial pressures to "standard" conditions as specified by contract or under guarantee, manufacturers use so-called "correction factors" which are based on actual tests or are calculated from the performance of the perfect engine using assumed or observed Rankine cycle efficiencies. These factors, of course, should be agreed upon by the parties interested, because they vary with type and design of engine, load, and initial and final steam conditions. For example, a manufacturer of a well-known line of four-valve high-speed non-condensing engines uses the following correction factors: Corrections in water rate at full load for varying initial pressures; 200 lb. deduct 5 per cent; 175 lb. deduct 3 per cent; 150 lb. ("standard," no correction); 125 lb. add 5 per cent; 100 lb. add 8 per cent; 75 lb. add 12 per cent.

Example 36. — A single-cylinder, single-valve, non-condensing counter-flow engine gave the following results under test conditions: Initial gage pressure, 100 lb. per sq. in.; initial quality, 99 per cent; atmospheric exhaust; water rate at rated load 30 lb. per i.hp-hr. Calculate the water rate under guaranteed conditions of 150 lb. initial pressure and dry steam, assuming that a reduced Rankine cycle efficiency of 10 per cent at the higher pressure was agreed upon.

Solution. — Actual Rankine cycle efficiency, equation (143)

$$\frac{2547}{W(H_i - H_n)} = \frac{2547}{30\ (1180 - 1030)} = 0.566.$$

The values for H_i and H_n corresponding to 100 lb. pressure and atmospheric exhaust may be conveniently taken from the Mollier diagram.

Rankine cycle efficiency at 150 lb. pressure, as per agreement.

$$= 0.566 - 10 \text{ per cent of } 0.566, \text{ or } 0.51 \text{ approximately.}$$

Water rate at 150 lb. pressure and dry steam may be calculated from equation (143), by substituting $E = 0.51$ (as per preceding calculation), $H_i = 1195$ (heat content of dry steam at 150 lb. gage pressure) and $H_n = 1016$ (heat content at atmospheric pressure after adiabatic expansion from initial conditions), thus:

$$0.51 = \frac{2547}{W\ (1195 - 1016)}$$

$W = 28$ lb. per i.hp-hr., water rate under "guarantee" conditions.

184. Increasing Rotative Speed. — High rotative speed does not necessarily mean high piston speed. An 8-in. by 10-in. engine running at 300 r.p.m. has a piston speed of only 500 ft. per min., whereas a 36-in. by 72-in. Corliss running at 60 r.p.m. has a piston speed of 720 ft. per min. The classification "high speed" and "low speed" refers to rotative speed only, the former above and the latter below, say 150 r.p.m.

On account of the reduction of thermodynamic wastes, a high-speed engine should give theoretically a higher efficiency than the same engine at a lower speed, all other conditions being the same. The effect of speed upon economy is decidedly marked in engines and pumps taking steam full stroke. For example, tests of a 12-in. by 7 1/4-in. by 12-in. simplex direct-acting steam pump at Armour Institute of Technology showed a steam consumption of 300 lb. per i.hp-hr. at 10 strokes per minute, and only 99 lb. at 100 strokes per minute.

Tests of engines using steam expansively, however, do not furnish conclusive evidence on this point, some showing a decided gain (Peabody, "Thermodynamics," p. 425), others little or no gain (Barrus, "Engine Tests," p. 260). For example, a small Willans engine showed an increase in economy of 20 per cent on increasing the rotative speed from 111 to 408 r.p.m. (Peabody, "Thermodynamics," p. 402), whereas the compound locomotive at the Louisiana Purchase Exposition showed a loss in economy for the higher speeds (Publication by the Pennsylvania Railroad Company). On the other hand, a comparison of the performances of high- and low-speed Corliss engines shows little difference in economy, and a general comparison between high- and low-speed engines furnishes little information, since nearly all high-speed engines are of a different class from the low-speed ones. High-speed engines are comparatively small in size, require larger clearance volume, and are usually fitted with a single valve. Rotative speed is limited by design, material, workmanship, and cost of subsequent maintenance. Speeds of 600 r.p.m. and more are not unusual with single-acting engines, whereas 300 r.p.m. is about the limit for double-acting machines with strokes over 12 in. in

length. A comparison of tests of high-speed and low-speed engines in this country, irrespective of design and construction, shows the former to be less economical than the latter in most cases. In Europe, high-speed engines are developed to a high degree of efficiency, and their performances are comparable with the best grade of low-speed engines.

High-speed engines as a class have the advantage of being more compact for a given power, are simple in construction and relatively low in first cost; on the other hand, they are subject to comparatively rapid depreciation, excessive vibration, and are usually less economical in steam consumption.

185. Decreasing Back Pressures. — In the non-condensing engine the minimum back pressure is that of the atmosphere (the Stumpf exhaust-ejector type engine excepted), and the back pressure in the condensing type is limited only by the degree of vacuum developed in the condenser. If pistons, valves, and stuffing boxes are tight and the steam passages are of ample area, the power developed by a given weight of steam will increase as the back pressure is decreased, or, for a given output, the water rate will decrease with the decrease in back pressure. The higher the ratio of expansion, the greater will be the effect of a given variation in back pressure. For this reason, non-condensing engines as a class are less influenced by variations in back pressure than are condensing engines by equivalent variations in vacua. For a given mean effective pressure, however, the influence is the same in both classes of service. With each type and grade of engine and initial steam conditions, there is a back pressure at which maximum economy is effected, but this critical point can only be determined by actual test. For example, a small high-speed non-condensing piston engine in the laboratories of the Armour Institute of Technology showed a minimum water rate at a vacuum of 22 in. (referred to 30-in. barometer) while the best performance of a small cross-compound Corliss was obtained with a 26-in. vacuum. Both engines were supplied with saturated steam at an initial pressure of 125 lb. abs. On the other hand, a small experimental engine designed for high initial pressures, vacuum, and superheat showed decreasing water rates up to the maximum vacuum obtainable, 28.6 in. (See Table 62.) Back pressures effecting minimum water rates are not necessarily the most economical from an overall standpoint, because the heat of the liquid at exhaust and the steam equivalent of the power required to produce the vacuum, to say nothing of the additional fixed and operating charges, have not been considered. If the condensate is discharged to waste, the water rate (neglecting fixed and operating charges for the condenser equipment) is a true measure of the heat economy, but if the heat of the exhaust is reclaimable the point of minimum heat consumption is not necessarily

coincident with that of the minimum water rate. This is clearly shown by the curves in Fig. 252, which, though strictly applicable to a specific case, are characteristic of counterflow engines in general. Referring to Fig. 252 it will be seen that the minimum water rate corresponds to a vacuum of 28 in. but the net heat supplied per unit output corresponds to a 21-in. vacuum. If the steam equivalent of the power required to operate the condenser equipment were deducted from the water rate or net heat supplied, the point of most economical performance would probably be at a still lower vacuum. Engines under 200 hp. rated capacity are seldom operated condensing, because the cost, fixed and operating, of producing the vacuum usually exceeds the gain in heat economy.

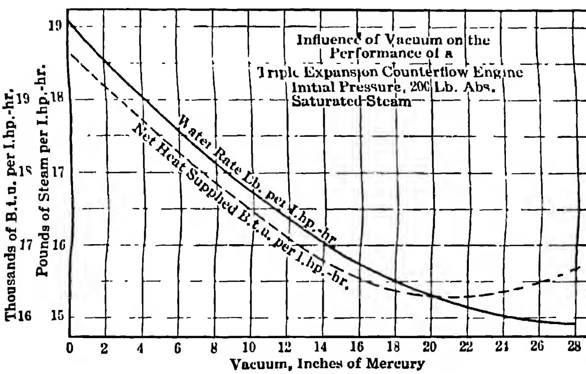


Fig. 252.

Where there is a large demand for exhaust steam for heating or other industrial purposes, the engines are generally of the non-condensing type; but where only a portion of the exhaust is required, the compound condensing engine is often the better investment. In the latter case, steam for heating is bled from the receiver. Single-cylinder condensing engines may be bled at any point during the forward stroke, but this is not common practice because of the added complication of the bleeder control. (See paragraph 212.) The reduction in water rate, neglecting the weight of steam required to produce the vacuum, which may range from 1 to 10 per cent of the main engine steam, varies with the type and size of engine load, reduction in back pressure, and initial steam conditions. Some idea of the influence of condensing on the water rate of small counterflow reciprocating engines may be gained from the data in Table 59. The curves in Fig. 271 show the performance of the uniflow engine when operating condensing and non-condensing, and those in *Fig. 267* show the comparative results of large cross-compound counterflow engines.

Corrections for back pressures may be made in a manner similar to those for initial pressures, as discussed in paragraph 183.

A manufacturer of a well-known line of high-speed four-cycle non-condensing engines gives the following correction factors for increasing back pressures:

Corrections for Increased Back Pressures at Full Load. Add to the water rate for each 1 lb. back pressure above atmosphere; 1 per cent at 200 lb. initial pressure; 1 1/4 per cent at 175 lb.; 1 1/2 per cent at 150 lb.; 2 per cent at 125 lb.; 2 1/2 per cent at 100 lb.; and 3 per cent at 75 lb.

Example 37. — A manufacturer of a line of simple Corliss engines guaranteed a water rate of 18 lb. per i.hp-hr. at rated load for initial gage pressure of 125 lb. per sq. in., dry steam at admission, and vacuum of 26 in., referred to 30-in. barometer. If the best vacuum obtainable under test conditions is 24 in., what would be the water rate in order to meet the guarantee? An increase of 5 per cent in Rankine cycle efficiency was agreed upon for the reduced vacuum.

Solution. — From steam tables, the initial heat content at 125 lb. gage pressure is found to be 1192.2 B.t.u. per lb. From the Mollier diagram, or by calculation, the heat content at the end of adiabatic expansion to a 26-in. vacuum is found to be 921 B.t.u. per lb. Substituting these values in equation (143), noting that $W = 18$, and solving for E, we have

$E = 2547 \div 18\ (1192.2 - 921) = 0.523$, Rankine cycle efficiency under guarantee conditions.

Rankine cycle ratio at 24-in. vacuum by agreement $= 0.523 + .05 \times 0.523 = 0.55$ approx.

Substituting this value for E in equation (143) and $954 = H_n$ for the final heat content corresponding to a 24-in. vacuum, we have

$$0.55 = 2547 \div W\ (1192.2 - 954)$$
$$W = 19.5 \text{ lb. per i.hp-hr.}$$

186. Superheating. — The theoretical gain due to the use of superheat is comparatively small, as will be seen from Table 60. Considering the additional expense of equipment and maintenance of superheating apparatus the ultimate gain would appear to be a negative quantity. Practically, however, the heat economy of the piston engine is greatly increased by superheating. This apparent anomaly is due to the fact that the theoretical engine is assumed to operate in a non-conducting cycle and no condensation takes place except in doing work, whereas in the actual mechanism the cylinder is far from being non-conducting and considerable initial condensation takes place. The reduction of cylinder condensation due to the use of superheated steam is the principal reason for the marked gain in economy of the actual engine. The greater the cylinder condensation, the larger is the saving possible. As a rough approximation,

the steam consumption is reduced about 1 per cent for every 10 deg. fahr. increase in superheat, but the actual value depends upon the type and size of engine and the initial condition of the steam. In American practice superheat corresponding to a total steam temperature of 600 to 650 deg. fahr. appears to be the limit of commercial economy, but in Europe temperatures as high as 850 deg. fahr. have been employed with apparent ultimate economy

TABLE 59

EXAMPLES OF THE EFFECT OF CONDENSING ON THE ECONOMY OF SMALL RECIPROCATING ENGINES

Reference Number	Non-Condensing			Condensing				Increase Due to Condensing	
	Initial Gage Pressure	Horse power Developed	Steam Consumption, Lb per Hp hr	Initial Gage Pressure	Back Pressure, Lb per Sq In Abs.	Horse-power Developed	Steam Consumption, Lb. per Hp-hr.	In Power, Per Cent	In Economy, Per Cent
1	147	51.7	19.2	119	1.6	83.4	14.8	52.5	25
2	148	510	19.3	147	4	...	16.9	*	12.5
3	126	83	23.8	130	7.4	116	19.1	39.8	19.7
4	67.6	209	28.9	67	4.5	213	22	1.9	23.5
5	103.8	177.5	22.1	103.8	1.2	155	16.5	*	25.1
6	114	160	31	114	.	168	27	2	12.9
7	96	120	23.9	96	4	145	19.4	20.8	18.8
8	118	267	23.21	119	1.2	276.9	16	3.7	31
9	75.9	310	25.6	79	6.4	336	20.5	8.7	19.9
10	62.5	451	30.1	63.6	7.8	444	23	*	23.6
11	186.7	40.1	18.7	184.6	1.6	29.8	12.7	*	32

* Cut-off changed for best economy

TABLE 60

THEORETICAL EFFICIENCIES AND WATER RATES

Rankine Cycle Superheated Steam

Initial Pressure 200 Lb per Sq In Abs

Superheat, Deg. Fahr.	Efficiency		Water Rate	
	Condensing*	Non-condensing	Condensing*	Non-condensing
0	31.88	18.60	6.94	13.44
50	32.03	18.71	6.72	12.96
100	32.24	18.92	6.52	12.49
150	32.49	19.18	6.34	12.03
200	32.77	19.51	6.16	11.57
250	33.09	19.89	5.98	11.10
350	33.81	20.76	5.67	10.20
400	34.20	21.25	5.48	9.77
500	35.04	22.12	5.16	9.00

* Absolute back pressure 0.5 lb. per sq. in.

So far as steam consumption per unit output is concerned, all engines of whatever type and size show greater economy with superheated steam than with saturated steam, and the economy increases with the superheat up to the maximum temperature at which the plant can be operated; but when the extra investment and operation costs are considered, the maximum commercial economy measured in dollars and cents usually occurs at a temperature considerably lower than this maximum.

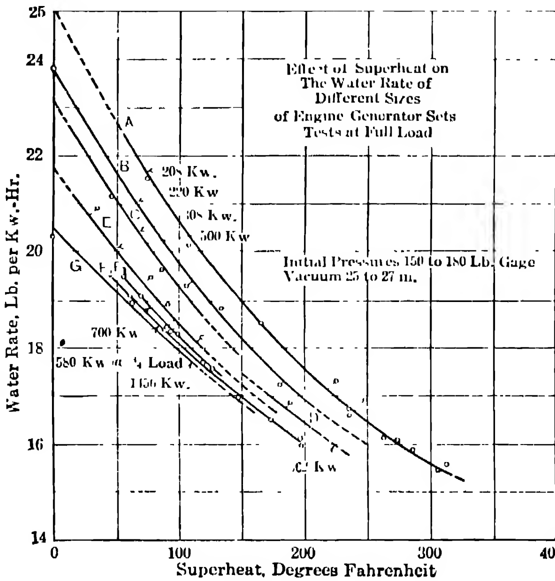


Fig. 253.

The higher the superheat, the less will be the influence of the size of engine on the water rate, and if sufficient superheat is put into the steam, all sizes of engines of given type and design will probably have the same water rate. This is illustrated by the curves in Fig. 253, which, though strictly applicable to the particular type of engines tested, are characteristic of engines in general.

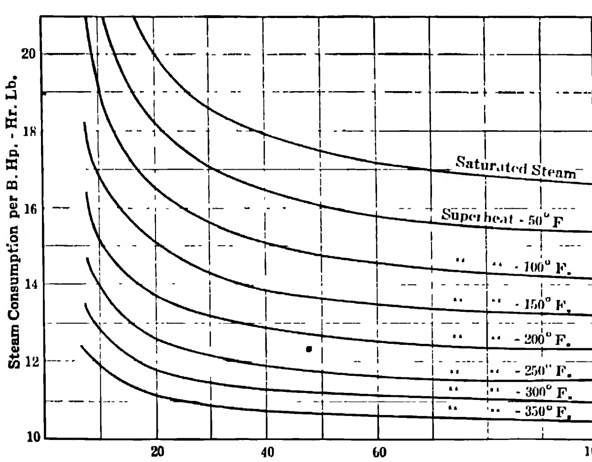


Fig. 254. Effect of Varying Superheat on Steam Consumption.

The higher the superheat, the less will be the influence of cylinder condensation on the water rate and the flatter will be the performance curve. This is shown by the curves in Fig. 254.

A few typical performance curves showing the influence of superheat on different types of counterflow engines are shown in Figs. 253 to 256. See Fig. 262 for influence of initial superheat on the performance of a uniflow engine.

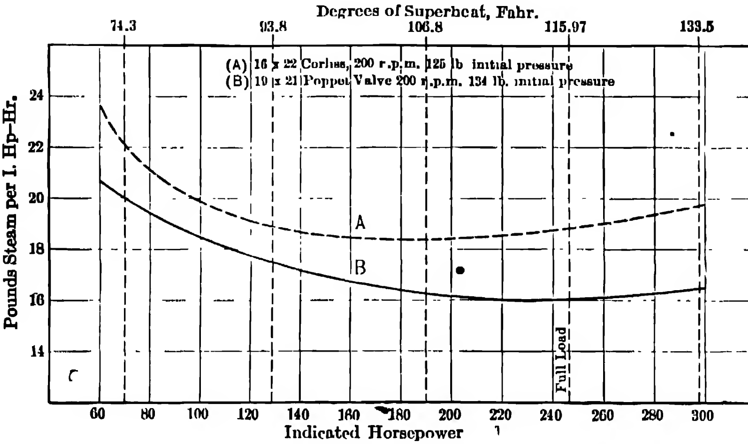


FIG. 255. Comparative Water Rates of a Corliss and a Poppet Four-valve, High-speed Engine.

187. Jacketing. — A few years ago it was common practice to make the walls of the cylinder double and fill the space with steam at boiler

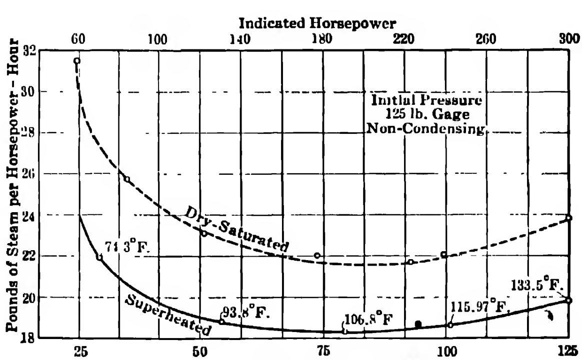


FIG. 256. Influence of Superheat on the Water Rate of a 16-in. by 22-in. "Ideal" Corliss Engine.

pressure. In some designs the cylinder heads were also jacketed. The function of the jacket is to reduce cylinder condensation with a view of reducing the surface losses. With certain types of engines there are

conditions under which the net heat supplied per unit output is less with the jackets in commission than when operating without them, while with others there is little or no improvement. The larger the area of inner surfaces exposed to the action of the working steam, the greater will be the benefits from jacketing. Increased speed, lower ratios of expansion, and larger units tend to reduce the surface losses and hence the effects of jacketing. High speed dampens the temperature fluctuation of the

TABLE 61

PERFORMANCE OF AN EXPERIMENTAL QUADRUPLE-EXPANSION PISTON ENGINE WITH HIGH INITIAL TEMPERATURES, SUPERHEAT, AND INTERMEDIATE SUPERHEATING

(Zeit d. Ver deut Ingr, July 2, 1921, p 718)

	H.P. Cyl.	I.P. Cyl (1)	I.P. Cyl (2)	L.P. Cyl.
Cylinder, diam. in.	5.3	9.5	11.2	26.8
Stroke, in.	15.7	15.7	23.6	23.6
Initial pressure, lb. abs.	793.8	246.9	57.3	11.3
Initial superheat, deg. fahr.	815	568	536	436
Exhaust temperature, deg fahr.	572	365	212	.
M.e.p., lb. per sq. in.	301.3	58.9	8.2	4.9
I.hp.	35.2	27.7	36.7	47.7
Water rate, lb. per i.hp-hr	21.5	27.3	20.6	15.8
Rankine cycle efficiency per cent	91	79.8	78.6	80.0

Total i.hp.	147.4	Vacuum, in. Hg.	28.6
Combined water rate, lb. per i.hp-hr.	5.12	Combined Rankine cyc. eff., per cent	81.7
Combined heat consumption, B.t.u. per i.hp-hr.	8197	Combined thermal eff., per cent	31

walls, low ratios of expansion increase the mean wall temperature, and the larger sizes have a higher ratio of volume to surface. For this reason large, high-speed, heavily loaded engines show the least gain from jacketing. Jacketing has the greatest effect in low-pressure cylinders, since the surface losses, temperature gradient, and the ratio by weight of jacket to working steam, are large. Tests conducted on triple-expansion engines with a view of ascertaining the influence of jacketing show that there is no gain in the high-pressure, very little in the intermediate, but a large gain in the low-pressure cylinders. Head jackets are usually more effective than cylinder jackets. In this country, jackets are seldom used except in connection with triple-expansion counterflow and single-cylinder uniflow engines, because better results may be obtained by initial superheating. A revival of the steam jacket for small single-cylinder counter-

flow engines has been stimulated by the Prosser "High-economy" engine in which the cylinder, heads, piston, ports, and valve chest are all jacketed with live steam. Figure 257 shows a longitudinal and Fig. 258 a trans-

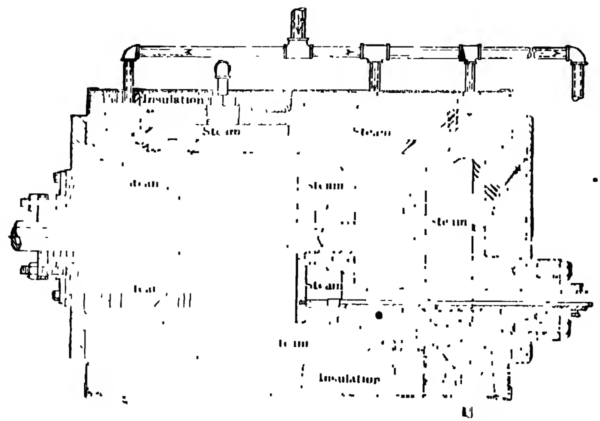


FIG. 257. Longitudinal Section through Cylinder of Prosser "High-economy" Engine.

verse section through the cylinder of this engine. Two plain steam-tight piston valves control the admission and the exhaust, each operated by its own eccentric on the engine shaft. In other respects the engine differs but little from any low-clearance high-speed automatic engine. Some idea of the exceptional economy of this design may be gained from the curves in Fig. 259 which were plotted from tests conducted at Purdue University on an experimental engine which was far from being mechanically correct. Recent tests on commercial units built by the Chandler and Taylor Co. have shown much better results, equaling if not surpassing the performance of uniflow engines of the same size and speed and for the same steam conditions. For influence of jackets on the water rate of uniflow engines, see Fig. 262.

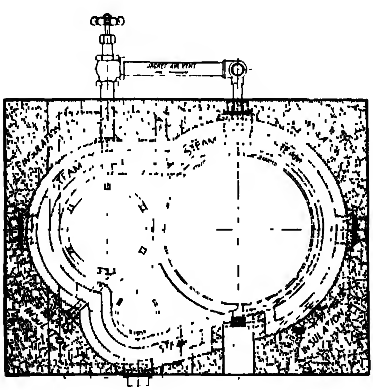


FIG. 258. Prosser "High-economy" Engine — Transverse Section through Cylinder.

188. Receiver Reheaters: Intermediate Reheating. — The receivers between the cylinders of multi-expansion engines are frequently equipped with heating coils, the function of which is to superheat the exhaust

steam before delivering it to the cylinder immediately following, with a view of reducing the losses occasioned by cylinder condensation. The coils are supplied with live steam under boiler pressure and may serve to evaporate a portion of the moisture or to actually superheat the steam supplied to the following cylinder. The question of the propriety of using reheaters for saturated steam is an open one, since reliable data relative to their use are meager and discordant. The conditions under which

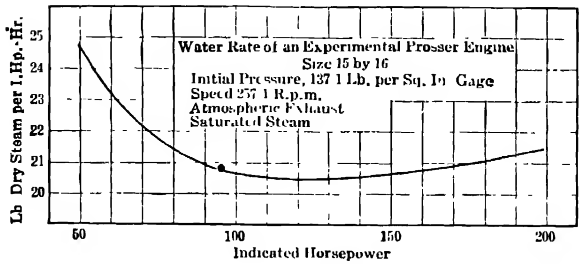


FIG. 259.

the few recorded tests were made are too diverse to warrant definite conclusions. Some show an appreciable gain in economy, others a decided loss. A reheater is of little value in improving the thermodynamic action of the engine, and is probably a loss unless it produces a superheat of at least 30 deg. fahr.; to be fully effective it should superheat above 100 deg. fahr. (L. S. Marks, *Trans. A.S.M.E.*, 25-500.) The effectiveness of the reheater will evidently be increased by the removal of the greater portion of the moisture from the exhaust steam before it enters the receiver. In the 5500-hp. engine at the Waterside Station in New York, it was shown that both jackets and reheaters, either together or alone, were practically valueless, throughout the working range of load. (*Power*, July, 1904, p. 424.) Many similar cases may be cited which show no gain in economy with the use of the reheaters. In all cases the reheater effects a great reduction in the condensation in the low-pressure cylinders, but the resulting gain, considering the condensation in the reheater coils, may be little, if any. On the other hand, with properly proportioned reheaters, the gain may be considerable, particularly with superheated steam. Practically all European engines operating with highly superheated steam are equipped with receiver-reheaters. Some idea of the exceptional economy effected with high initial pressure and intermediate superheating may be gained from the data in Tables 61 and 62. The values in Table 62 are based upon tests of a 11.1-in. and 26.5-in. diameter by 23.4-in. stroke tandem-compound experimental engine (*Żeit. d. Ver. deut. Ingr.*, July 2, 1921). As pressures advance,

the reheating cycle becomes more advantageous, as is evidenced by the latest steam-turbine projects in which initial pressures of 1200 lb. per sq. in. are contemplated with reheating between the high- and low-pressure units. In the locomobile types of engine plant, the intermediate reheating is effected by heating coils placed in the path of the furnace gases, and, in the latest steam-turbine plant, superheaters of the duplex type are to be employed for reheating purposes.

In triple-expansion pumping engines, receiver-reheaters are found to effect an appreciable gain in economy, and practically all such engines are equipped with them. In electric power plants where the load is a widely fluctuating one, the reheater has been virtually abandoned. Apart from the consideration of fuel economy, all tests show a marked increase in the indicated power of the low-pressure cylinder (5 to 15 per cent) and to that extent it increases the capacity of the entire engine.

189. Compounding. — If the entire expansion, instead of being effected in a single cylinder, is allowed to take place in two or more cylinders, the engine is said to be "compounded." The term "compound" without qualification, however, refers only to the two-cylinder arrangement. If expansion takes place in three stages the engine is known as a triple-expansion engine; similarly, the four-stage machine is called a quadruple-expansion engine. When high-pressure steam is admitted into a single engine of the ordinary double-flow type and expansion is carried down to a comparatively low point, a large portion is condensed by the metal surfaces; at the end of the stroke and during exhaust, some of the water is re-evaporated, but the steam so formed is discharged without doing useful work. If the same weight of steam is expanded through the same pressure range in a multi-expansion engine, the temperature range in each cylinder will be less, initial condensation will be reduced, and part of the heat lost in the first cylinder by leakage and clearance will do work in the second cylinder, and so on throughout each stage. The higher the temperature range the more pronounced will be the thermal economy effected by compounding. The number of stages is limited commercially because of the first cost, complexity, cost of lubrication, attendance, and maintenance.

Cylinder ratios for high-speed, single-valve, compound engines vary from about 1 to 2 1/2 with 100 lb. pressure to about 1 to 3 with a pressure of 150 lb., and for low-speed condensing engines from 1 to 3 with 125 lb. pressure to about 1 to 4 with a pressure of 175 lb. G. I. Rockwood recommends a ratio as high as 1 to 7, and a number of engines designed along this line have shown exceptional economy. For variable-load operation, two stages appear to give the best ultimate economy. In case of very large condensing engines, the last stage consists of two cylinders because

of the unwieldy and costly size of a single unit. For constant loads, as in pumping stations and large marine installations, three and four stages appear to be the best investment. The ratio of expansion for a multi-expansion engine is the ratio of the volume at release in the low-pressure to that at cut-off in the high-pressure cylinder. Commercially, it is usually taken to be the ratio of the volume of large to small cylinder divided by the fraction of the stroke at cut-off in the high-pressure cylinder. For example, a compound engine with cylinders 24-in., 48-in. by 48-in., cutting off at 1/3 in the high-pressure cylinder, has a nominal ratio of expansion of $4 \div 1/3 = 12$. The number of expansions at rated load in multi-expansion condensing engines varies widely, ranging from 10 to 33, with an average not far from 16.

TABLE 62

TANDEM-COMPOUND ENGINE, OPERATING WITH SUPERHEATED STEAM AND INTER-STAGE SUPERHEATING, WITH LARGE RATIO OF EXPANSION IN L.P. CYLINDER

Date of Test	Mar. 9, 1921	Feb 25, 1921	Mar. 2, 1921	Apr. 2, 1921
Revolutions per minute	146.0	145.6	145.2	145.6
Mean effective pressure, h.p. cyl., lb.	17.1	19.2	19.1	19.1
Mean effective pressure, l.p. cyl., lb.	5.45	5.14	5.38	5.26
Indicated horsepower, h.p. cyl.	50.2	53.3	51.6	52.3
Indicated horsepower, l.p. cyl.	52.2	49.0	51.0	50.1
Total indicated horsepower	102.2	102.3	102.6	102.4
Initial steam pressure, lb. abs.	110.2	110.2	117.6	122.0
Pressure entering l.p. cyl., lb. abs.	13.9	16.1	18.3	19.8
Vacuum, inches of mercury	28.3	26.5	26.0	25.6
Initial steam temperature, deg. fahr.	608	600	617	581
Final steam temp., h.p. cyl., deg. fahr.	266	268	293	268
Initial steam temp., l.p. cyl., deg. fahr.	413	413	428	425
Hourly total steam consumption, lb.	815.1	884.4	895.4	916.3
Steam consumption per i.hp-hr.	7.98	8.64	8.73	8.93
Total heat consumption per i.hp-hr., B.t.u.	11,175	12,078	12,236	12,414
Rankine efficiency, h.p. cyl., per cent	78	81.2	78.5	82.3
Rankine efficiency, l.p. cyl., per cent	80	81.2	85.0	81.8
Rankine efficiency of engine, per cent	80	83	84	84.2

The respective advantages and disadvantages of compounding may be tabulated as follows:

ADVANTAGES

1. Possibility of high range of expansion.
2. Decreased cylinder condensation.
3. Decreased clearance and leakage losses.
4. Equalized crank effort.
5. Increased economy in steam consumption.

DISADVANTAGES

1. Increased first cost due to multiplication of parts.
2. Increased bulk.
3. Increased complexity.
4. Increased wear and tear.
5. Increased radiation loss.

190. Uniflow Cylinders. — By placing the exhaust ports midway of the length of the cylinder, as shown in Fig. 260, the steam will be forced to flow through the cylinder in one direction only. Combining this feature with high-grade steam-tight inlet valves, minimum clearance volume, and steam-jacketed heads, we have the principle characteristics of the modern uniflow engine, which, because of its remarkable heat economy, is rapidly supplanting the counterflow type for practically all classes of service where low water rates are important factors. As will be seen from Fig. 260, the double-acting straight uniflow engine has two steam valves, one at each end for admission of steam only, and no exhaust valves, the piston itself performing this function. The piston is long, practically 9/10 of the stroke, and the cylinder therefore is longer than that of a counterflow engine of the same diameter and stroke. The exhaust ports have an area approximately three times that of any other type of engine, so that wire drawing is reduced to a minimum. Steam enters the cylinder

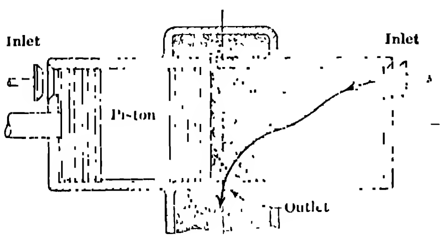


FIG. 260. Principles of the Uniflow Cylinder.

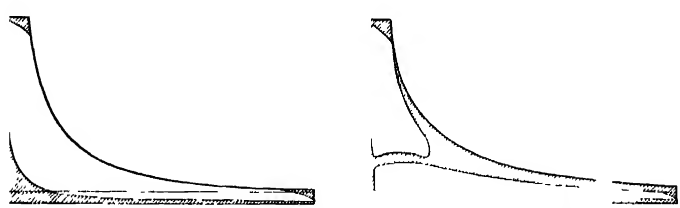


FIG. 261. Comparison of Indicator Cards for the Same Load and Steam Conditions.

after it has passed through the head jacket, forces the piston to the end of its stroke, and discharges through the central exhaust ports. Due to the jacketing effect there is little or no change of temperature in the cylinder up to cut-off, after which expansion takes place with a consequent drop in temperature. This drop in temperature continues until exhaust takes place. The exhaust does not sweep over the cylinder head and other surfaces at the inlet end as in the counterflow engine; hence the surfaces are not cooled to the same extent. When the piston uncovers the ports, any water of condensation is quickly swept from the cylinder, and the steam trapped in the cylinder at the beginning of the return stroke

is practically dry. The heat of compression is therefore not absorbed in re-evaporating moisture as in the case of the counterflow cylinder, but is imparted to the steam, so that with the added heat from the heat jacket the compressor line is substantially adiabatic. This reduction of the surface losses enables a single uniflow cylinder to operate with as high a degree of expansion as compound or triple expansion counterflow cylinders, and with the same or even better heat economy. See Fig. 261. Head jackets are essential under all conditions of steam, but cylinder jackets are seldom used except with saturated or slightly superheated steam at low mean effective pressures. The influence of cylinder jackets on the performance of a condensing uniflow engine at varying steam tempera-

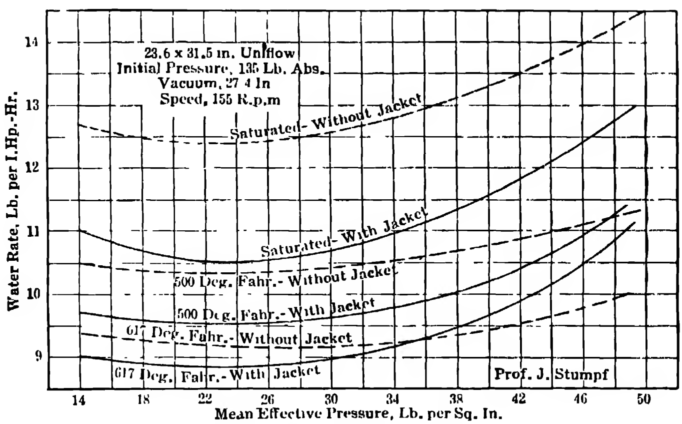


FIG. 262. Influence of Cylinder Jackets on the Water Rate of a Uniflow Engine with Varying Superheat.

tures is shown in Fig. 262. It will be seen that the value of the cylinder jacket decreases with the increase in superheat and mean effective pressure and eventually results in an actual heat loss.

The compression in the straight uniflow cylinder begins at about 10 per cent and continues during the remaining 90 per cent of the return stroke. The beginning of compression is fixed and cannot be altered. The final pressure depends therefore entirely upon the amount of clearance space into which the steam is compressed and the pressure at the beginning of compression. For maximum economy, compression should be carried to practically initial pressure; therefore, an engine designed for 200 lb. pressure would not operate satisfactorily if the pressure were reduced to 150 lb. for the reason that the amount of compression is constant with fixed clearance volume and exhaust pressure. The straight uniflow engine is primarily intended for condensing service, because the weight

of steam entrapped at the beginning of compression at low condenser pressure is so small that the final pressure at the end of compression is less than the initial, even with minimum practical clearance. When, however, the uniflow engine is required to operate non-condensing or with high back pressure, the pressure at the end of compression becomes excessive unless provision is made for preventing it from doing so. Excessive compression may be prevented by increasing the clearance volume, delaying the beginning of compression through the use of auxiliary ports, and by withdrawing a portion of the steam when a certain predetermined pressure is reached. Applications of these principles to various designs of American uniflow engines, together with performance data, will be found in paragraph 197.

191. Binary Vapors. — The efficiency of any heat engine may be increased by extending the range of heat availability of the working fluid. In practice, the range is limited by the pressure-temperature relationship of the working fluid. For each fluid there is a pressure and temperature range beyond which it is inpractical to go, because of the physicial limitations of the materials employed in generating heat at the higher level, and of the physical properties of the working fluid and cooling media at the lower level. Mercury, for example, boils under atmospheric pressure at 677 deg. fahr., and under a vacuum of 28 in. at 455 deg. fahr. The corresponding temperatures for water are 212 deg. fahr. and 101 deg. fahr. respectively, and for methyl alcohol 151 deg. fahr., and 50 deg. fahr. respectively. By employing, say, three cylinders, each operating through a complete cycle with a different fluid, and connected in such a manner that the condenser for the first fluid is the boiler for the second, and so on, high thermal efficiencies may be effected with comparatively low pressure ranges. The earliest attempts were made with steam as the high-temperature fluid and ether or sulphur dioxide as the low-temperature fluid. While remarkable results were obtained from this combination compared with those of the steam engine of that day, they were no better than the performance of the modern high-grade uniflow engine. Binary vapor engines of this class are not found in modern practice, because the added complexity of the plant and increased first cost offset any thermal gain except possibly in connection with the poorest design of steam engine.

An experimental binary-vapor plant, designed by W. L. R. Emmet, in which mercury is used for the high-temperature and steam for the low-temperature stage, gives promise of high commercial heat economy, but sufficient data are not available to show whether or not this combination unit can compete successfully with the modern single-vapor plant.

Binary-vapor Engines: Jour. Frank. Inst., June, 1903; Elec. World and Engrg. Aug. 10, 1901; Engineer, U. S., Aug. 1, 1903; Sibley Jour. of Engrg., March, 1902.

The Emmet Mercury-steam Plant: Power, Aug. 3, 1920, p. 167. Power Plant Engrg., Jan. 1, 1924, p. 97; Mech. Engrg., March, 1924, p. 235.

192. High-Speed Single-valve Simple Engines. — This style of engine is made in sizes varying from 10 to 500 hp. The cylinder dimensions vary from 4-in. by 5-in. to 24-in. by 24-in. and the rotative speed from 400 to 175 r.p.m.

When ground is limited or costly and a large percentage of the exhaust steam is necessary for heating or manufacturing purposes, the high-speed non-condensing engine is suitable for horsepowers of 200 or less, being compact, simple in construction and operation, and low in first cost. For sizes larger than this, the compound or uniflow engine may prove a better investment, except where fuel is very cheap or large quantities of exhaust steam are to be used for manufacturing purposes during the greater part of the year.

Small high-speed engines are seldom operated condensing, since the gain due to reduction of back pressure is more than offset by the extra cost of the condenser and appurtenances.

Engines are ordinarily rated at about 75 per cent of their maximum output. For example, a 12-in. by 12-in. non-condensing engine running at 300 r.p.m. with initial steam pressure of 80 lb. per sq. in. gage is normally rated at 70 hp., though it is capable of developing 90 hp. at the same speed.

The steam consumption of high-speed single-valve non-condensing engines at full load ranges from 26 to 50 lb. per i.hp-hr., depending upon the size of the unit and the conditions of operation. An average for good practice is not far from 30 lb. With superheated steam a steam consumption as low as 18 lb. per hp-hr. has been recorded.

Figure 263 shows the steam consumption of a number of single-valve high-speed engines at various loads. The steam consumption is fairly constant from 50 per cent of the rated load to 25 per cent overload, but for lighter loads the economy drops off rapidly. The desirability of operating the engine near its rated load is at once apparent. The curves show a marked economy in favor of the larger cylinders, but the engines are not of the same make, and the conditions of operation are somewhat different.

The most economical cut-off for a simple engine, for the steam conditions usually employed, is about one-third to one-fourth stroke when running non-condensing, and about one-sixth when running condensing.

While much higher performances are recorded for well-designed high-speed single-valve engines, it is not advisable to count on a better satura-

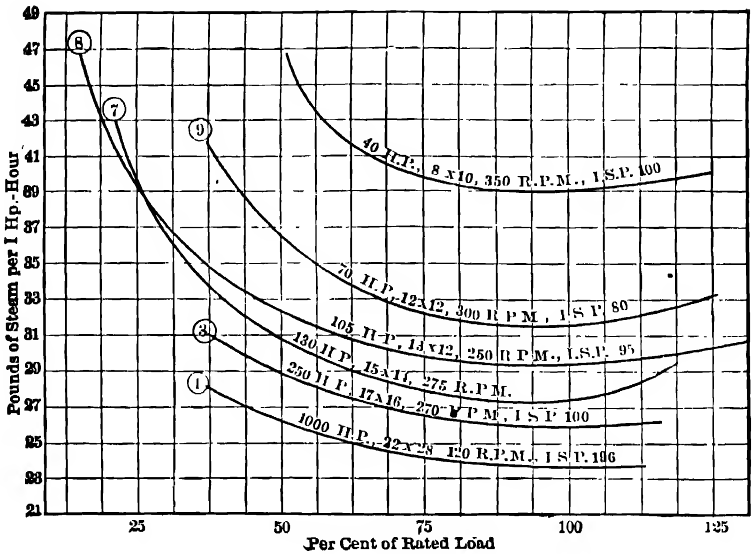


FIG. 263. Typical Economy Curves of High-speed Single-valve Non-condensing Engines. Saturated Steam.

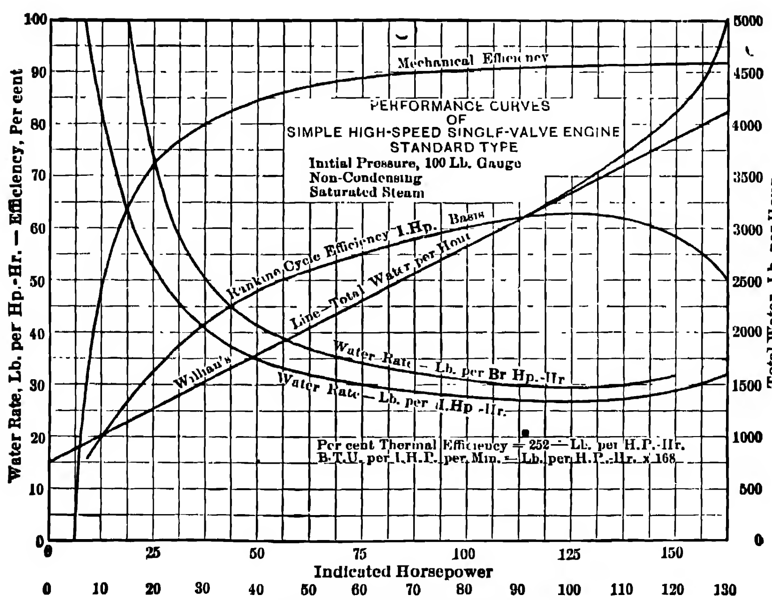


FIG. 264.

ted steam consumption for this type than 30 to 35 lb. of steam per i.hp-hr. for initial pressure of 100 lb. gage or less.

The curves in Fig. 264 give the performance of a modern, high-grade, unjacketed, 15-in. by 14-in. high-speed, single-valve, simple, non-condensing engine at various ratings. It is not likely that this type and size of engine can be designed to better materially the results shown in the curves for the given conditions.

In general, when the requirements for exhaust steam are in excess of the steam consumption of a simple non-condensing engine, a high-grade economical engine is without purpose.

193. High-speed Multi-valve Simple Engines. — The steam distribution in a single-valve engine may give good economy for a very small range in load but may be far from satisfactory for a wide range. This must necessarily be so, since admission, cut-off, release, and compression are all functions of one valve, and any change in one results in a change of the others. To obviate the limitations of the single valve, many builders design engines with two or more valves. With a two-valve engine, cut-off is independent of the other events, and with four valves all events are independently adjustable. In addition to the flexibility of the valve gear, the chief feature of the four-valve engine lies in the reduction of the clearance volume, which is made possible by placing the valves directly over the ports. The valves may be of the common slide-valve, or of the rotary type. As a class, four-valve engines are more economical than those having a smaller number of valves. The advantages and disadvantages of the four-valve over the single-valve engines may be tabulated as below:

Advantages	Disadvantages
1. Better steam distribution.	1. Increased number of parts.
2. Better regulation	2. Increased first cost.
3. Reduced clearance volume.	3. Requires greater attention.
4. Less valve leakage.	
5. Better economy.	

The steam consumption of a high-speed Corliss non-condensing engine at full load varies from 24 to 27 lb. of saturated steam per i.hp-hr. (pressure 125–140 lb. gage) with an average not far from 25 lb. With moderate superheat, the water rate may run as low as 17 lb. per i.hp-hr. The poppet-valve type appears to be more economical in steam consumption than the Corliss, and a water rate for saturated steam as low as 18.9 lb. per i.hp-hr. has been recorded. A very high degree of superheat can be used with the poppet-valve type, and water rates as low as 16 lb. per i.hp-hr. (initial pressure 150 lb. gage, superheat, 250 deg. fahr.) are not

unusual. The high-speed four-valve engine is usually operated non-condensing. Rankine cycle efficiencies over 80 per cent have been realized with both saturated and superheated steam. An exceptional record for a condensing unit is reported by Lentz. With steam at 461 lb. abs. initial pressure and steam temperature of 1018 deg. fahr., a 100-hp. Lentz unjacketed simple engine developed an indicated horsepower on a steam consumption of 5.67 lb. per hr.

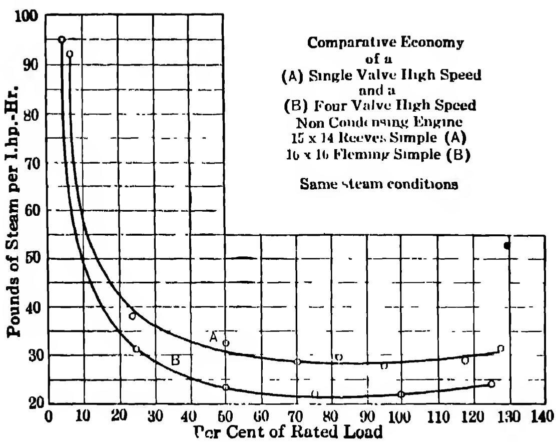


FIG. 265.

Figure 265 gives a comparison between a single-valve and a four-valve (Corliss type) high-speed engine, using saturated steam, and though the engines differ slightly in size, the conditions of operation were comparable and the marked gain in economy of the latter over the former is apparent. Both performances are exceptional, and a 10 to 15 per cent greater steam consumption may be expected in average good practice.

As a general rule, single-valve simple engines do not exceed 400 hp. in size, whereas 1000 hp. is not an uncommon size for the multi-valve type.

194. Medium and Low-speed Multi-valve Simple Engine. — A comparison of tests of high- and low-speed single-valve engines irrespective of design and construction shows the former as a class to be less economical than the latter. With four-valve engines there is no such disparity, and the high-speed type has shown just as good economy as the low-speed class.

Of the various types of simple, low- or medium-speed, four-valve engines, the poppet-valve appears to be the more economical in heat consumption, but so much depends upon the grade of workmanship that general comparisons are apt to lead to error. A comparison of the steam consumption of a high-speed, four-valve Corliss and a four-valve poppet engine, non-condensing, is shown in Fig. 255. As the size and initial pressure are somewhat in favor of the poppet-valve mechanism, the results are not strictly comparable, but the exceptional economy of both types is apparent from the curves.

195. Compound Engines. — It should be borne in mind that the principal object of compounding is to permit the advantageous use of

high pressures and large ratios of expansion, and consequently this type of engine need not be considered for pressures lower than 125 lb. per sq. in. gage. This does not signify that 125 lb. is the limiting pressure for compounding; on the contrary, compound condensing engines with initial pressures as low as 90 lb. have shown better heat economy than simple engines of the same capacity, but the thermal gain for these low pressures is usually more than offset by fixed charges and other practical considerations. In general, compounding increases the steam economy at rated load from 10 to 25 per cent for non-condensing engines and from 15 to 40 per cent for condensing engines. Compound engines range in size from the 100-hp. tandem, single-valve, automatic, high-speed, non-condensing unit to multi-valve, cross-compound condensing units of 4000 hp. or more. Compound engines have been built up to 10,000 hp. rated capacity, but the steam turbine has practically superseded the piston engine for sizes larger than 2000 hp. for electric power generation. Non-condensing high-grade compound engines of the full poppet-valve type, with superheated steam, and the uniflow engine are more economical in steam consumption than non-condensing steam turbines of the same capacity; but first cost, size, and attendance are decidedly in favor of the turbine, at least for sizes over 2000 hp. Low rotative speed and reversibility, however, are points in favor of the engine, but the former may be offset by the turbine in connection with suitable reduction gearing.

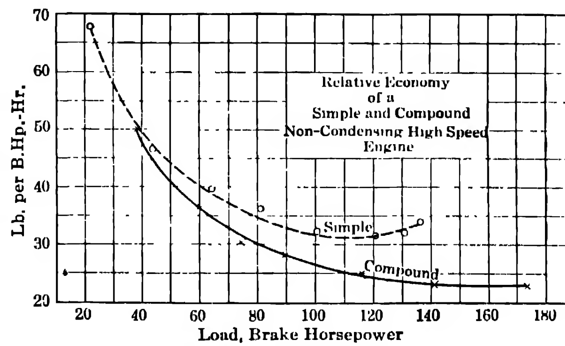


FIG. 266. Comparison of a Simple and Compound Single-valve Engine.

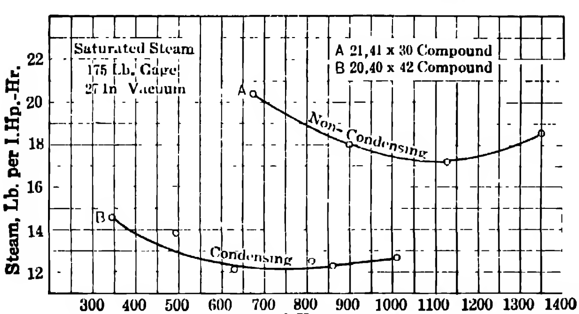


FIG. 267. Performance of a Cross-compound Engine—Condensing vs. Non-condensing.

With saturated steam under the conditions found in general practice,

the water rate of the standard type of single-valve compound non-condensing engine ranges from 22 to 27 lb. per i.hp-hr. at rated load. Since this type of engine as ordinarily constructed permits of only a moderate amount of superheat, the water rate with superheated steam is seldom less than 20 lb. per i.hp-hr. Condensing under a standard vacuum of 26 in. reduces the water rate approximately 20 per cent.

The four-valve, compound, non-condensing engine has a full-load water rate, with saturated steam, ranging from 17 to 22 lb. per i.hp-hr., and with superheated steam an economy as low as 12 lb. per i.hp-hr. has been recorded. Rankine cycle efficiencies as high as 83 per cent for saturated steam and 90 per cent for superheated steam, have been realized.

So much depends upon the initial pressure, degree of vacuum, and initial temperature that general figures for condensing practice are without purpose. With saturated steam the best performances are in the neighborhood of 75 per cent of the theoretical Rankine cycle efficiency, while, with highly superheated steam, 85 per cent of the Rankine cycle efficiency has been realized.

196. Triple- and Quadruple-Expansion Engines. — With the exception of the vertical triple-expansion pumping engine, compound engines having more than two stages are obsolete so far as American practice is concerned. There is no question but that multi-cylinder compound engines can be built which will give better water rates than the two-cylinder design, in fact, the highest efficiency so far recorded for any steam prime mover is that of a small Schmidt experimental quadruple-expansion engine; but heat economy is only one of the factors entering into the total cost of energy. The more cylinders, the larger will be the unit, the more complex the mechanism, and the higher the first cost and cost of maintenance. For electric power generation, the steam turbine has superseded the piston engine for large sizes, and the single-cylinder uniflow and the two-cylinder compound counterflow have taken the place of the multi-cylinder compound for units under 4000 hp. While the vertical triple-expansion pumping engine has held first place for thirty-five years as the ideal pumping engine for large water works because of its high heat economy, reliability, and low upkeep, it is being rapidly replaced by the turbine-driven, geared, centrifugal pump. The latter occupies a cubical space of approximately one-fifth to one-sixth that of the former and weighs about one-tenth as much for the same capacity. The reduced first cost of equipment, buildings, and foundations, and the saving in space usually offset the thermal gain of the reciprocating unit. High initial pressures and temperatures with intermediate superheating are favorable to the multi-cylinder compound engines, but it is not likely that more than two cylinders will be employed in the immediate future.

Some idea of the exceptional performances of vertical triple-expansion pumping engines may be gained from the data in Table 63.

TABLE 63

PERFORMANCE OF TYPICAL TRIPLE-EXPANSION PUMPING ENGINES

Date of Test	Type	Location	Rated Capacity Millions of U. S. Gallons	Initial Gage Pressure	Initial Superheat Fahr	Duty		Lb. Steam per I Hp-Hr
						Per Thousand Lb. of Steam	Per Million B t u.	
1–15–10	Allis	Milwaukee, Wis	12	124.6	Sat.	175.4	151.0	10.82
5– 2–00	Allis	Boston, Mass	30	185.5	Sat.	178.5	163.9	10.33
2– 4–06	Allis	St. Louis, Mo	20	140.6	Sat.	181.3	158.8	10.66
4–29–10	Holly	Albany, N. Y	12	153.0	Sat.	182.1	. .	. . . . .
3–10–10	Holly	Frankfort, Pa	20	180.2	Sat.	184.4	. .	. . . .
5– 2–09	Holly	Louisville, Ky	24	155.1	109	195.0	164.5	9.46
11–15–18	Holly	Cleveland, O	10	199.3	102	201.6	169.7	8.96
4–23–14	Holly	St. Louis, Mo.	20	159.4	102	202.6	166.7	9.77*
10–14–18	Allis-C.	Cleveland, O.	20	206.3	130	211.5	188.7	8.89

Date of Test	Type	R.p.m.	Water Actually Pumped Millions of U. S. Gal. 24 Hr.	Net Head Pumped Against Lb. per Sq. In.	Indicated Horsepower	Water Horsepower	Thermal Efficiency Per Cent
1–15–10	Allis	20.4	12.130	121.0	673.0	618.0	20.25
5– 2–00	Allis	17.7	30.314	61.0	801.5	747.8	21.63
2– 4–06	Allis	16.5	20.070	104.0	859.2	839.6	20.92
4–29–10	Holly	22.3	12.193	139.5	. . . . . .	726.0	. . . .
3–10–10	Holly	20.1	21.219	95.7	. . . .	817.0	. . . . .
5– 2–09	Holly	24.0	24.111	90.0	925.7	879.4	22.54
11–15–18	Holly	27.1	10.010	382.9†	737.3	672.7	21.81
4–23–14	Holly	20.0	20.610	297.7†		1074.9	21.40*
10–14–18	Allis-C.	21.0	20.380	377.7†	1417.0	1343.0	24.27

* Water Horsepower Basis. † Feet.

197. The Uniflow Engine. — The uniflow engine is rapidly replacing the wasteful single-cylinder counterflow engine and also the compound engine which is economical in steam consumption over a narrow range of load only and otherwise undesirable on account of the comparatively large amount of floor space that is required for its installation. This type of engine adapts itself to the large majority of conditions under which counterflow engines are used, condensing and non-condensing, high or moderate (but not low) steam pressures with or without superheat, high or low speed, and belted or direct connected. Uniflow engines are especially well adapted for driving rolling mills, blowing engines, textile mills, and crusher plants, in addition to electric generators where an economical

and reliable type of prime mover of moderate size is desired. The uniflow engine contains considerable more material than a simple Corliss engine of the same power; the cylinder must be larger in diameter because of the low mean effective pressure, and longer because of the extra length of piston. Furthermore, for best economy, initial pressures and temperatures are considerably higher than those commonly used with the ordinary single-cylinder counterflow engine; hence the first cost is 15 to 25 per cent greater. The steam consumption, however, is lower than with any type of single-cylinder counterflow engine and equal to or even better than that of the best compound. The water-rate curve is flat, thus insuring good economy over a wide range in load. The single-cylinder counterflow engine has a normal cut-off at 20 to 30 per cent of the stroke, while the uniflow has its best economy and is rated at a cut-off ranging from 8 to 10 per cent; and, since the governor permits of a cut-off as late as 60 to 70 per cent, it is evident that heavy overloads may be carried when necessary. Uniflow engines are built with piston, Corliss, or poppet valves, though the great majority of American designs have poppet valves.

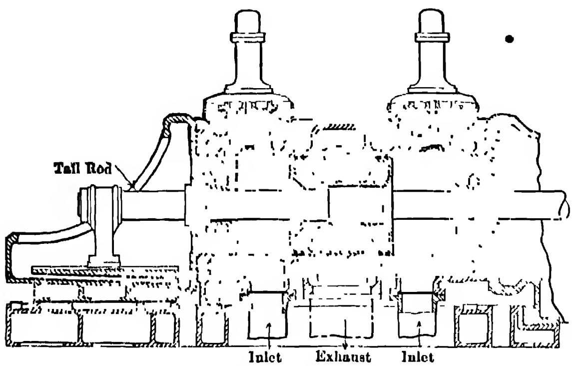


FIG. 268. Mesta Heavy-duty Condensing Uniflow Engine.

In basic principle, all American-built uniflow engines for condensing service are identical with the standardized Stumpf design and differ only in details of governor, cylinder arrangement, and valve construction. For non-condensing service, however, the cylinders are usually equipped with auxiliary exhaust valves or other devices, so as to prevent excessive compression. The use of these valves tends to neutralize the effect of the uniflow principle, but careful tests have demonstrated that non-condensing engines thus equipped show a materially lower water rate than straight counterflow engines. A few well-known designs will be briefly described with a view of bringing out the different methods adopted for preventing excessive compression.

Figure 268 shows a section through the cylinder of a **Mesta** condensing uniflow engine, illustrating the true uniflow principle as advocated by Prof. J. Stumpf. This particular design is constructed in single-cylinder

units of 500 to 600 hp., and twin-cylinder units up to 1200 hp. The inlet valves are of the resilient double-seated poppet-valve type, actuated by a non-releasing gear. The latter consists of a roller and cam driven by an eccentric on a lay shaft. Provision for releasing excessive compression in case the vacuum fails is effected by two *artificial* clearance valves, opening into auxiliary clearance spaces, one at each end of the cylinder. These valves are automatically operated by a pilot valve which is connected to the condenser as well as to the live steam line. The clearance valves automatically open when the vacuum drops, and automatically close when the vacuum is re-established. By means of this control of clearance, the engine can be changed from condensing to non-condensing, or *vice versa*, without interrupting the operation of the engine. The heavy pistons are of the full "floating type;" that is, they are supported by the piston rod which is extended as indicated.

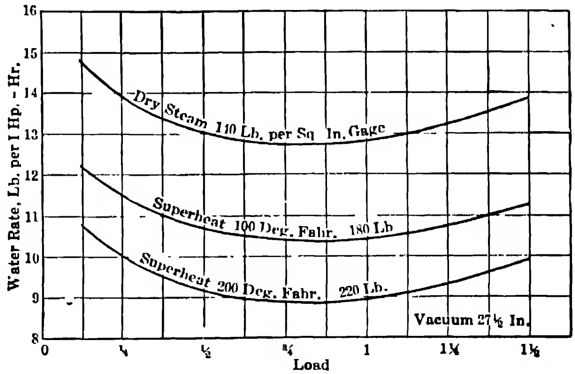


FIG. 269. Performance of a Mesta Heavy-duty Uniflow Engine.

Figure 270 shows a section through the cylinder of a "**Universal**" uniflow engine intended primarily for non-condensing service, but which automatically adjusts itself to condensing service, and *vice versa*, should occasion demand. It will be seen from the illustration that in addition to the regulation inlet valves and central exhaust ports, two auxiliary ports are placed between the cylinder heads and the central port, in such a manner that compression will not occur until the piston closes these

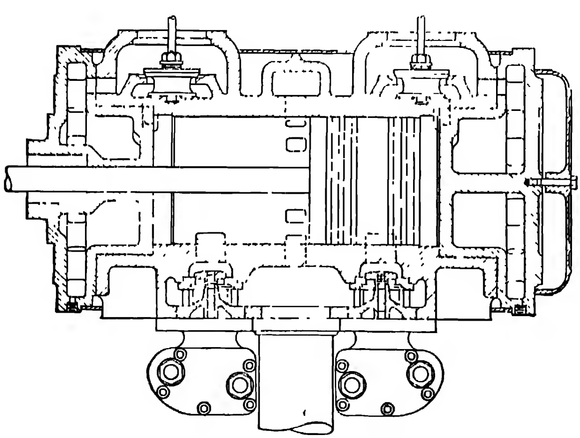


FIG. 270. Section through Cylinder of a "Universal" Uniflow Engine.

ports. The auxiliary ports are opened and closed by mechanically operated single-beat poppet valves, which in turn are controlled by a valve gear driven by a separate eccentric. When it is desired to run condensing, the exhaust valves are made inactive and the engine in operation becomes a true uniflow engine. The operation for non-condensing service is as follows: Steam enters the cylinder through the double-beat inlet valves and is exhausted through the central port in the same manner as when operating condensing. On the return stroke, part of the vapor

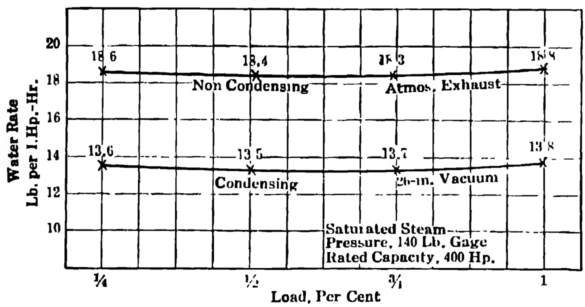


FIG. 271. Water Rate of a 21- by 22-in. "Universal" Uniflow Engine.

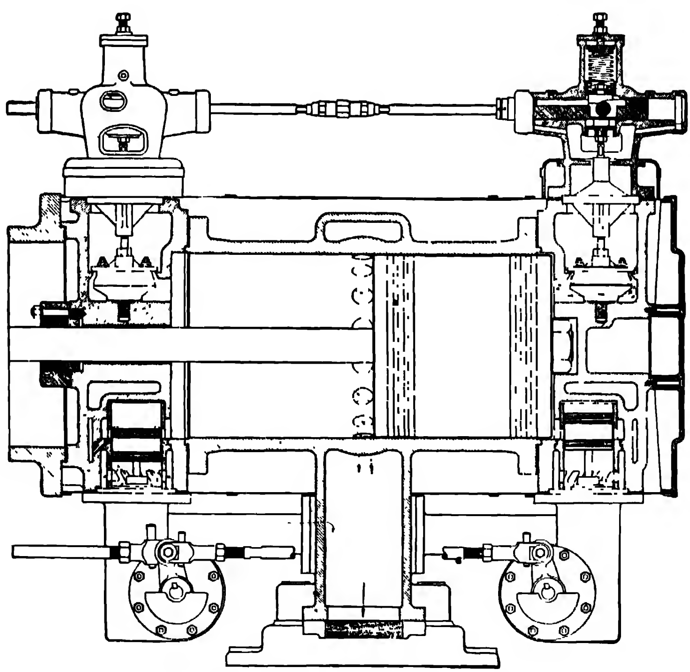


FIG. 272. Murray Uniflow Engine.

is forced through the auxiliary ports into the main exhaust pipe, until the piston covers the ports, when compression begins as in any counterflow en-

gine. The auxiliary exhaust valves remain open during compression, but since the piston covers the port openings there is no escape of steam. Compression is controlled solely by the location of the auxiliary ports and not by the closing of the auxiliary valves. The clearance volume in this design is somewhat larger than if there were no auxiliary port openings, but the difference is small.

In the **Murray** non-condensing uniflow engine, Fig. 274, the auxiliary valves are placed at the ends of the cylinder. These valves are mechanically operated and may be adjusted so that compression will begin at any

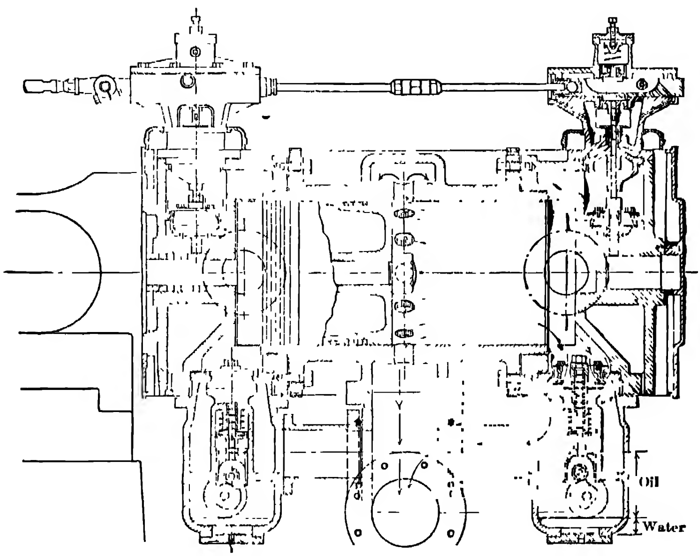


FIG. 273. Ames "Controlled-Compression" Uniflow Engine.

predetermined period during the return stroke. It will be seen that, during the return stroke of the piston and up to the point when the auxiliary valve closes, the steam cycle is practically that of a counterflow engine. This is true of all uniflow engines employing auxiliary exhaust valves at the ends of the cylinder, but the reduction in economy on this account is very small as is evidenced by actual test results.

In the **Ames "Controlled-Compression"** uniflow engine, the auxiliary valves are located as shown in Fig. 273. The valves are of the double-beat poppet type.

The **Harrisburgh** non-condensing uniflow engine, Fig. 274, is of the single piston-valve type in which excessive compression is controlled by providing "dual" clearance volume. Steam is expanded from a small clearance and discharged through the central exhaust ports in the manner

of all uniflow engines. On the return stroke, the steam entrapped by the piston is compressed first into a large chamber and finally, as the piston nears the end of the stroke, into a small clearance. The steam which has been compressed in the large chamber enters, by automatic action of the

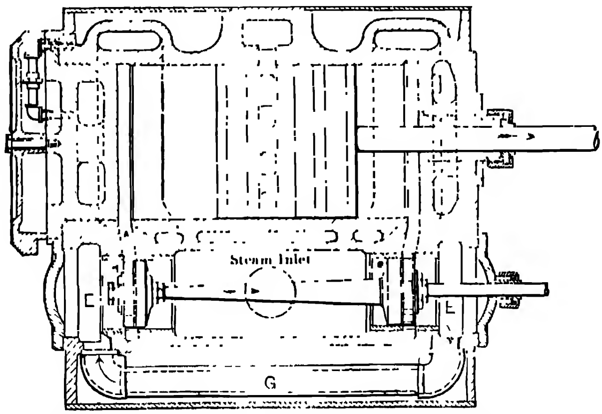


FIG. 274. Harrisburgh "Dual-Clearance" Uniflow Engine.

valve, the cylinder at the opposite end and, mixing with the expanding steam on the return stroke, expands with it and passes out the central ports to the exhaust. It will be seen that by this action none of the exhaust vapor is by-passed to waste, as is the case with auxiliary exhaust ports. Figure 274 shows the outer edge of the valve opening the port at

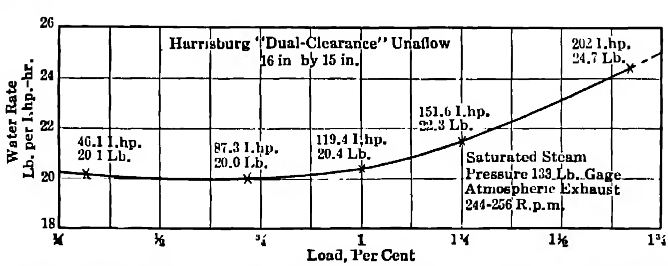


FIG. 275. Water Rate of a Harrisburgh "Dual-Clearance" Uniflow Engine.

the head end, thus permitting steam from the auxiliary clearance chambers *E*, *G*, *E*, to pass into the cylinder and mix with the expanding steam. The opposite end of the valve has closed the port and compression occurs in the cylinder and cylinder clearance only.

In the **Chuse** uniflow engine the valves are of the single-beat poppet type. The auxiliary exhaust valves for non-condensing operation are also of this type.

198. The Locomobile. — Although classified under "steam engines" the term "locomobile" applies to the complete power plant and not to the engine only. In Europe this type of plant has been developed to a high degree of efficiency, and with very high superheat steam consumptions as low as 6.95 lb. per i.hp-hr. have been recorded, corresponding to a coal consumption of 0.75 lb. coal per brake hp-hr. The locomobile is not much in evidence in American steam power plant practice.

Figure 276 shows a longitudinal section through a typical locomobile plant. The entire plant is self-contained and requires very little floor space. The engine, of the compound center crank type, is set upon the

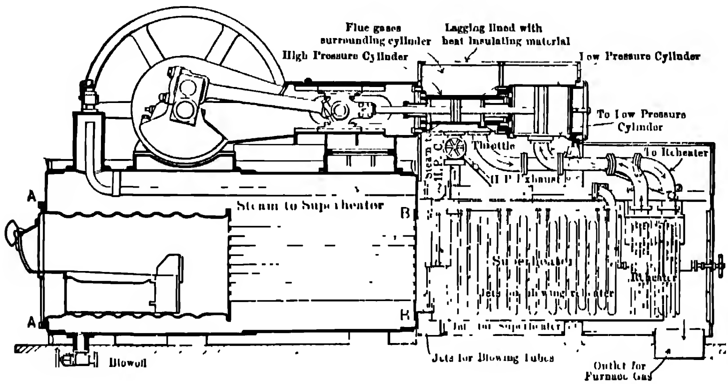


Fig. 276. Section through a Typical Locomobile Plant.

boiler with the cylinders projecting into the "smoke-box" so as to minimize piping and radiation losses. Steam is generated in an internally fired tubular boiler at a pressure of 225–275 lb. per sq. in. gage and is superheated to a final temperature of 600–700 deg. fahr. Exhaust steam from the high-pressure cylinder is reheated by an auxiliary superheater, adjoining the main superheater, before it enters the low-pressure cylinder. The feedwater is heated by an economizer or reheater placed in the breeching. The condenser is of the jet type and is provided with a rotary air pump. In Europe the locomobile is built in various sizes ranging from 50 to 2000 hp.

199. Rotary Engines. — The rotary engine differs from the reciprocating engine in that the piston, or equivalent, rotates about the cylinder axis. Its operation is entirely different from that of the steam turbine; in the rotary engine the static pressure of the steam actuates the piston, and in the turbine the momentum of the steam is imparted to the rotating element.

Over 2200 patents have been issued to date on rotary engines, but

not a single machine has yet been able to compete with the reciprocating engine as regards steam economy. The advantages of the rotary engine are many, and for this reason innumerable inventors have been exerting their skill in the development of this type of prime mover; but unfortunately the impracticability of satisfactorily packing the rubbing surfaces has more than offset the advantages, and the commercially successful machine is yet to be found.

The writer has tested out various types of rotary steam engines, and the best has been but a poor competitor of the ordinary grade of reciprocating mechanism.

200. Selection of Type. — Modern operating conditions are so diversified and at the same time so specialized that the selection of the type of piston engine best suited for a proposed installation is an increasingly difficult problem. That engineers are not agreed as to the best practice is evidenced by the different types of engines selected for practically identical operating conditions. General rules are without purpose, since each particular installation is a problem in itself. Floor space, capacity, rotative speed, cost of fuel, water rate, steam pressure, water supply, load characteristics, exhaust steam requirements, size of foundation, vibration, first cost, attendance, and maintenance all govern the selection of type. The principal factor governing the size of units is the station load curve, or rather, load curves. When these load curves are known, the problem is a comparatively simple one, but when they must be assumed, as is generally the case with a new project, it is largely a matter of experience.

Because of its compactness, low first cost, simplicity, and low maintenance costs, the single-valve, single-cylinder, high-speed, non-condensing engine is usually the better investment in situations where the individual units do not exceed, say, 200 hp. and where the larger portion of the exhaust can be used for heating or other purposes. If, however, considerable quantities of exhaust steam are discharged to waste, during the non-heating season, the single-cylinder, high-speed, four-valve, non-condensing engine, or the non-condensing uniflow with auxiliary exhaust valves, or the equivalent, may be the more economical. If the load during the non-exhaust-utilizing period is fairly constant and somewhat near its economical rating, the four-valve counterflow engine is preferable to the uniflow because of its lower first cost and economy in space and foundation requirements. If, however, the load fluctuates within wide limits, the flat water-rate curve of the uniflow may offset the advantage in first cost and space requirements of the four-valve design. Uniflow engines require massive foundations and this may prove to be a serious objection in a small plant. Single-valve, single-cylinder engines, as ordinarily

constructed, are not suitable for steam temperatures over 450 deg. fahr., while standard designs of the uniflow type are designed for temperatures as high as 600 deg. fahr. Single-cylinder, high-speed, counterflow engines are seldom operated condensing because the cost of the condensing system is excessive compared with the increase in heat economy.

For non-condensing service and for sizes ranging from 200 to 1000 hp., the choice lies between the single-cylinder, low-speed Corliss, the compound counterflow, and the single-cylinder uniflow. For low initial pressures and where the speed limitations permit, the simple Corliss is a good selection, but for high initial pressures the compound counterflow and the simple uniflow are much more economical in steam consumption. The water rate curve of the uniflow is flatter than either of the others and therefore operates to advantage where the load departs considerably from normal rating. The non-releasing Corliss engine is basically a low-speed machine and therefore requires a very heavy and expensive generator for direct connected service. This calls for additional floor space and foundations and increases the size and cost of the building. For pressures over 125 lb. per sq. in., preference should be given to the compound counterflow and the uniflow engine.

Condensing units over 1000 hp. rated capacity are usually of the Corliss type where the initial pressure does not exceed 100 lb. gage. For higher pressures the compound counterflow or single uniflow are ordinarily the better investment. Compressors, and hoisting and rolling mill engines are usually of the compound counterflow type, though the single-cylinder uniflow is finding increasing favor with many engineers. While the piston type of engine is still used for driving electric generators over 500 kw. rated capacity, the majority of new installations for this service are equipped with turbo-generators. This is due not so much to heat economy, because the piston-type engine can be designed to equal if not exceed the performance of any steam turbine, as it is to the saving effected in first cost and cost of attendance.

In case of alternating-current machinery — and the great majority of turbo-generators are of this type — it must be remembered that the r.p.m. of the generator depends upon the number of poles used in the machine and the frequency of the current to be generated; thus

$$\text{Frequency or cycles} = \frac{\text{r.p.m.} \times \text{no. of poles}}{120} \qquad (152)$$

The greater number of poles increases the diameter, which necessarily increases the material and cost of construction. Low-speed machines, therefore, cost much more than machines of higher speed and equal capacity. Turbo-generators are inherently high-speed machines and

this accounts in a large measure for their adoption in central station practice.

Selection of Steam Engines: Power, Jan. 16, 1923, p. 96; May 13, 1924, p. 766.

PROBLEMS

1. A 40-hp. non-condensing piston engine uses 500 lb. of saturated steam per hour when running idle and 1600 lb. per hour when operating at full load; initial pressure 115 lb. abs. Draw the unit water-rate curve, assuming that the total water rate follows the "Willans" straight-line law.

2. A 15-inch by 18-inch poppet-valve engine uses 18.8 lb. steam per i.hp-hr. at rated load, initial pressure 145 lb abs.; back pressure 0 lb. gage; initial quality 99 per cent; release pressure 4 lb. gage; mechanical efficiency at rated load 91 per cent Required (on both i.hp. and br.hp. basis):

a. Heat consumption per hp-hr.

b. Thermal efficiency, per cent.

c. Rankine cycle ratio, per cent.

d. Cylinder efficiency, per cent.

3. The Rankine cycle ratio of a compound poppet-valve engine is 90 per cent at full load; initial pressure, 150 lb. abs.; temperature of steam at admission, 450 deg. fahr.; back pressure 16.1 lb abs. Calculate the full-load water rate, lb per i.hp-hr.

4. If the exhaust from the engine in Problem 3 is used for heating purposes, required the full-load water rate, lb. per i.hp-hr chargeable to power.

5. A simple engine indicates 160 hp. on a dry steam consumption of 31 lb. per i.hp-hr.; initial pressure 130 lb. abs, back pressure 0 lb. gage. By shortening the cut-off, and by reducing the back pressure to 4-in. mercury (referred to a 30-in. barometer) the water rate is reduced to 22 lb. per i.hp-hr., the load remaining the same. If the condensing equipment requires 10 per cent of the steam supplied to the engine for its operation, required the net gain or loss in heat consumption per i hp-hr. due to condensing

6. Which is the more economical from a heat consumption standpoint, a simple non-condensing engine using 26 lb. dry steam per i hp-hr., initial pressure 100 lb. abs., or a compound condensing engine using 12 lb. steam per i.hp-hr., initial pressure 290 lb. abs., superheat 350 deg. fahr., back pressure 2-in. mercury? Which is the more perfect of the two?

7. A 600-hp uniflow engine uses 11.0 lb. of steam per i.hp-hr. when operating at full load; initial pressure 175 lb. gage, superheat 150 deg. fahr, vacuum 26 in. If the Rankine cycle ratio increases 1.2 per cent for each 1-in. decrease in vacuum from 26 in. up to atmospheric pressure and decreases 1.5, 4 and 10 per cent for increase in vacuum from 26 to 27, 28 and 28.5 in., respectively, required the most economical vacuum on the net-heat supplied basis, assuming that the condensate is fed to the boiler at a temperature corresponding to the vacuum.

8. A non-condensing engine uses 22.4 lb of steam per i.hp-hr. under the following conditions: Initial pressure, 150 lb gage, superheat, 50 deg fahr., back pressure, 17 lb. abs. It is proposed to operate this engine condensing under a 26-in. vacuum, 100 deg. superheat and initial pressure, 125 lb. gage. If the Rankine cycle ratio is increased 5 per cent by the reduction in initial pressure, 5 per cent by the increased superheat, and decreased 24 per cent by the reduction in back pressure, required the water rate under the changed conditions. Increase and decrease in Rankine cycle ratios referred to Rankine cycle ratio under non-condensing conditions.

CHAPTER XI

STEAM TURBINES

201. General. — In 1896 the steam turbine as a practical machine was almost unheard of. To-day it is the most important prime mover in the power world, at least insofar as the large central station is concerned. For certain classes of service, such as steel rolling mills, hoisting, reciprocating compressors, and small non-condensing electric generating plants, the piston engine is usually the better investment, but even this field is being encroached upon by the geared turbine and the variable-speed and reversing motor driven by turbo-generators. While the piston engine will no doubt continue to be an important factor in power generation, it has been practically eliminated from consideration in large central stations. No radical changes have been made in the design of steam turbines during the past few years, though high steam pressures and temperatures have necessitated many modifications in structural details. As a heat engine, the steam turbine has been developed to a high degree of perfection, but considerable work of improvement remains to be done to successfully safeguard reliability of operation. Single-cylinder units have been constructed in various sizes ranging from a small non-condensing auxiliary drive rated at less than 1 hp. to large turbo-alternators of 50,000 kw. rated capacity. Multi-cylinder units of 70,000 kw. maximum capacity have been installed in a number of central stations.

The theory and design of the steam turbine is fully covered in many excellent text books on that subject and no attempt will be made to discuss this phase of the subject except in a very elementary manner. A few basic types have been described in detail, more with the object of bringing out the principles involved than for purposes of design.

A general classification of steam turbines is unsatisfactory because of the overlapping of the various groups, and the following chart is offered merely as a guide in arranging a few well-known turbines according to the fundamental principles involved in their operation.

In conformity with the practice of most manufacturers, turbines have been divided into three general classes, (1) impulse, (2) reaction, and (3) combined impulse and reaction. Strictly speaking, however, all turbines depend more or less upon both impulse and reaction for their operation, and the more suitable terms, velocity and pressure, have been proposed.

Steam Turbines.	Impulse.	Single Velocity.	De Laval " Class A."	Single-pressure Stage.
		Multi-velocity Stage.	Kerr. Terry. Sturtevant. Curtis. Westinghouse.	
		Single-velocity Stage.	Ridgway. De Laval. Rateau. Large Curtis.	Multi-pressure Stage.
		Multi-velocity Stage.	Curtis. Kerr.	
	Reaction.	Single-velocity Stage.	Allis-Chalmers.	
	Combined Impulse and Reaction.		Westinghouse.	

Impulse Type. — In the impulse type the steam is expanded in a stationary nozzle or group of nozzles, and the heat given up by the pressure drop imparts velocity to the jet itself. The jet impinges against the vanes or buckets on a rotating wheel and gives up its kinetic energy to the wheel. The steam pressure is the same on both sides of the vanes or buckets. If the entire pressure drop takes place in one set of nozzles and the resulting jet is directed against a single wheel, the turbine is classified with the **single-stage single-velocity group.** The velocity of the jet is very high, from 2000 to 4000 ft. per sec., and for satisfactory economy the peripheral velocity of the wheel must also be very high, from 700 to 1400 ft. per sec. The De Laval " Class A " turbine, the only example of this group, is no longer manufactured though a number are still in operation.

If the entire pressure drop takes place in a single set of nozzles and a single wheel is to be used at a comparatively low speed, satisfactory economy may be effected by **compounding the velocity.** That is, the jet issuing from the nozzle at a very high velocity is reflected back and forth from the vanes on the rotor to a series of fixed reversing buckets until all of the available kinetic energy of the jet has been imparted to the wheel. The steam pressure is the same on both sides of all vanes or buckets. The Terry single-stage turbine is representative of this group.

Low peripheral velocity and high efficiency may be obtained by **pressure compounding**; that is, expansion takes place in a series of successive nozzles instead of one nozzle. Only a part of the available heat energy

is converted into kinetic energy in each set of nozzles. For each set of fixed nozzles there is a corresponding rotor. This class of turbine, frequently called the Rateau type, is substantially a series of single-velocity impulse turbines placed side by side. The steam pressure in each stage is less than that in the preceding stage. The Kerr and large-sized Curtis turbines are representative of this group.

By **compounding both velocity and pressure** we have the multi-velocity and pressure type of which the Curtis turbine is the best-known example.

Reaction Type. — In the reaction type the conversion of potential to kinetic energy takes place in the moving blades as well as in the fixed blades. Only a small portion of the heat energy imparts velocity in the first set of fixed blades or nozzles. The jet issuing from this set of nozzles impinges against the first set of moving blades at a velocity substantially that of the moving blades, so that it enters them without impulse. The moving blades are proportioned so that partial expansion takes place within them and the resulting increase in velocity exerts a **reaction** upon the moving blades. The expansion is very gradual and a large number of alternately fixed and revolving blades are necessary to effect complete expansion. Because of the small pressure drop in each stage (seldom exceeding 3 lb. at any one row of blades), low peripheral velocities are possible with high overall efficiencies. Because of the number of stages necessary to effect complete expansion and the excess leakage over the blade tips in the high-pressure stages, the straight reaction principle is not used in turbines under 1000 kw. rated capacity. The Allis-Chalmers turbine is of this type.

Combined Impulse and Reaction Type. — In this class the high-pressure elements are of the impulse type and the low-pressure elements of the reaction type. The Westinghouse single-cylinder high-pressure condensing turbine is typical of this class and is virtually a combination of the Curtis and Parsons designs. Several European impulse turbines as recently designed are fitted with reaction blades adjacent to the nozzles, showing the tendency to merge the different fundamental types.

Turbines may be classified according to the service for which they are intended, as (1) **high-pressure non-condensing,** (2) **high-pressure condensing,** (3) **low-pressure,** (4) **mixed-pressure,** (5) **bleeder.**

Turbines may also be classified according to the direction in which the steam flows with reference to the rotor, as (*a*) **axial,** (*b*) **radial,** (*c*) **tangential.**

Turbines may be still further classified according to method of driving, as (1) **direct connected,** and (2) **geared;** or according to the number of cylinders and their arrangement, as (*a*) **single-cylinder,** (*b*) **multi-cylinder,** (*c*) **tandem-compound,** and (*d*) **cross-compound.**

Each of these types, with the exception of the radial-flow, is discussed later on in the chapter. There are no American turbines of the radial-flow type.

202. General Elementary Theory. — A given weight of steam at a given pressure and temperature occupies a certain known volume and contains a known amount of heat energy. If the steam is permitted to expand to a lower pressure, it is capable of doing a certain amount of work which, theoretically, will be the same whether the expansion takes place in the cylinder of a reciprocating piston engine, a rotary piston engine, or the nozzles and blades of a steam turbine.

Let W = rate of flow of the steam, lb. per sec.,
E = energy given up by 1 lb. of steam in expanding from the higher to the lower pressure, ft-lb.,
H_1 = initial heat content of the steam, B.t.u. per lb.,
H_n = final heat content of the steam, B.t.u. per lb.

Then the heat drop, or heat available for doing useful work, is $W(H_1 - H_n)$ B.t.u. per sec.

If the steam expands from an initial condition H_1 to a final state of H_n, the energy E_1, available for doing work is

$$E_1 = 777.5\ W(H_1 - H_n), \text{ ft-lb. per sec.} \tag{153}$$

In the ideal or perfect piston or rotary engine, all of this energy is imparted to the piston or equivalent and only an insignificant portion is utilized in imparting velocity to the steam itself.

If, instead of acting directly on the piston of a reciprocating or rotary engine, the entire expansion takes place in a frictionless nozzle or the equivalent, then the heat drop will impart velocity to the steam itself and the kinetic energy, E_2, developed by the jet will be

$$E_2 = WV_1^2 \div 2g, \text{ ft-lb. per sec.} \tag{154}$$

in which

V_1 = velocity of the jet in the direction of motion of the wheel as it issues from the nozzle, ft. per sec.

Now, if this jet is directed against the blades, vanes, or buckets of a turbine wheel, the force exerted by the jet against the vanes is WV_1/g lb. If the jet leaves the vanes at velocity V_n ft. per sec., it will exert a force in the direction of motion of $-WV_n/g$ lb. The sign of V_n is negative because its direction is opposite to that of V_1. The total force, P, measured in lb., acting on the vanes in the direction of motion is the algebraic

difference of the entering and leaving force or

$$P = WV_1/g - (-WV_n/g)$$
$$= W(V_1 + V_n) \div g \tag{155}$$

In the purely impulse turbine, the jet leaves the vanes at zero velocity; hence WV_n/g is zero and the force exerted by the jet on the vane in the direction of motion is

$$P = WV_1/g \tag{156}$$

The work, E_3, absorbed by the vanes, assuming no losses, is the product of the peripheral force, P, and peripheral velocity, u (ft. per sec.), or

$$E_3 = Pu = Wu\ (V_1 + V_n) \div g \tag{157}$$

For maximum theoretical efficiency the peripheral velocity must be one-half that of the jet or $u = 1/2\ V$, or $u = 1/2\ (V_1 - V_n)$ if only part of the energy is absorbed. Substituting these values for u in equation (157) and reducing, we have

$$E_3 = W(V_1^2 - V_n^2) \div 2g \tag{158}$$

In the purely impulse turbine $V_n = 0$; therefore, the work absorbed E_4, is

$$E_4 = WV_1^2 \div 2g \tag{159}$$

In the reaction turbine the entire heat drop does not take place in the fixed or stationary nozzles, but part occurs in the fixed nozzles and the remainder in the moving vanes; that is, the moving vanes are in reality nozzles expanding steam in much the same manner as the fixed vanes or nozzles.

If v_1 and v_n are the respective inlet and outlet velocities of the moving vanes in the direction of motion relative to the moving vanes, it can be shown that the force, P_1, acting on the moving vanes in the direction of motion, is

$$P_1 = W(v_n - v_1) \div g \tag{160}$$

and the work, E_5, absorbed by the moving vanes is

$$E_5 = W(v_n^2 - v_1^2) \div 2g \tag{161}$$

In the purely reaction turbine the jet from the stationary nozzles enters the moving vanes at the same velocity as the latter, or v_1 = zero, hence the work absorbed, E_6, is

$$E_6 = Wv_n^2 \div 2g \tag{162}$$

If the moving vanes are considered as stationary, then for a given heat drop $v_n = V_1$ and $E_4 = E_6$. In other words, the work done by a given heat drop is the same whether the expansion takes place in fixed or moving nozzles.

But $E_1 = E_4$; hence from equations (153) and (159)

$$777.5\ W\ (H_1 - H_n) = WV_1^2 \div 2g$$

from which

$$V_1 = 223.7\ \sqrt{H_1 - H_n}.[1] \tag{163}$$

A glance at equation (159) will show that if the entire heat drop takes place in a single nozzle or set of nozzles, very high jet velocities will result, and if a single set of vanes is employed to absorb the energy of the jet, the peripheral velocity of the rotor must also be high. In order to obtain high efficiencies and at the same time relatively low peripheral velocities, the turbine may be **staged** or **compounded**; that is, (1) the heat drop may take place by degrees in a number of nozzles (pressure compounding), (2) the kinetic energy of the jet may be absorbed by a series of alternate fixed and moving vanes (velocity compounding), or (3) a combination of pressure and velocity stages may be employed (pressure and velocity compounding).

If there are n pressure stages only, the theoretical **stage velocity,** V_s, assuming equal heat drops in each stage, is

$$V_s = 223.7\ \sqrt{(H_1 - H_n) \div n} \tag{164}$$

For maximum theoretical efficiency the peripheral velocity of the rotor, V_p, is one-half the stage velocity (see equation (182))
or

$$V_p = V_s \div 2 \tag{165}$$

If there are n' velocity stages only, then the peripheral velocity for maximum theoretical efficiency, V_p', is

$$V'_p = V_1 \div 2n' \tag{166}$$

Combining equations (163) to (166) and reducing, and making $n = n'$, we have

$$V_p = V_p'\ \sqrt{n} \tag{167}$$

That is, for the same heat drop and number of stages, lower peripheral velocities may be obtained by velocity compounding than by pressure compounding.

The heat supplied, heat converted to work, theoretical water rates,

[1] For most purposes it is sufficiently accurate to make 223.7 = 224.

and efficiencies for the various cycles employed are the same as for the reciprocating engine. These quantities are defined and analyzed in Chapter XXIII and hence need not be duplicated here.

Equations (153) to (167) are general and are applicable to all turbines of whatever make.

Heat Drop in Steam Turbines· Trans. A.S.M.E., Vol. 33, p. 325, 1911; The Engr., Mar. 8, 1912; U. S. Bureau of Standards, Reprint No. 167, 1911.

203. Single-pressure, Single-velocity-stage Impulse Turbine. — This is the simplest type of steam turbine and consists essentially of a single wheel revolving in a single casing fitted with one or more nozzles. Steam is completely expanded in the nozzle or nozzles (the number depending upon the size of the turbine) from the initial to the existing back pressure, and the kinetic energy of the jet is absorbed by a single row of vanes or buckets mounted on the periphery of the wheel. Since the total heat drop takes place in the nozzles, the velocity of the jet is very high and ranges from 2000 to 4000 ft. per sec. depending upon the initial and final steam conditions. For maximum efficiency the peripheral velocity of the wheel must be approximately half the effective velocity of the jet, or 1000 to 2000 ft. per sec. For the small wheels employed in this type of machine, this is equivalent to 20,000 to 40,000 r.p.m. Such rotative speeds are suitable only for very high-speed apparatus and some sort of reduction gearing is necessary if the turbine is to drive at lower speeds. If the wheel is driven at speeds less than approximately half that of the jet, the steam will leave the vanes with high residual velocity and considerable energy will be wasted. While a great many turbines of this class are still in use they are no longer manufactured primarily because of the high rotative speeds.

The **De Laval " Class A "** turbine is the best-known application of the single-pressure, single-velocity, impulse principle. A section through the casing and wheel is shown in Fig. 277. The rotor or wheel consists of a high-carbon steel disc fitted with a single row of drop-forged steel blades, and mounted on a light flexible shaft. A flexible shaft is employed because it is mechanically impossible to establish perfect rotative balance at very high speeds. The flexible shaft permits the wheel to " gyrate " about its center of gravity instead of being forced to rotate about its geometrical center as would be the case if a rigid construction were used. The casing is of cast steel and encloses the wheel. The nozzles are inserted in the casing as shown in detail in Fig. 278. The blades are made with a bulb shank and fitted into slots milled in the rim of the wheel. The flanges at the outer end of the blades are brought into contact with each other and calked so as to form a continuous ring. The governor is of the centrifugal type and controls the speed by throttling the steam supply.

The operation of the turbine is as follows: Steam enters the steam chest, Fig. 277 and Fig. 278, through the governor valve and is distributed to the various adjustable nozzles, varying in number from 1 to 15 according to the size of turbine. In the earlier types the nozzles were uniformly

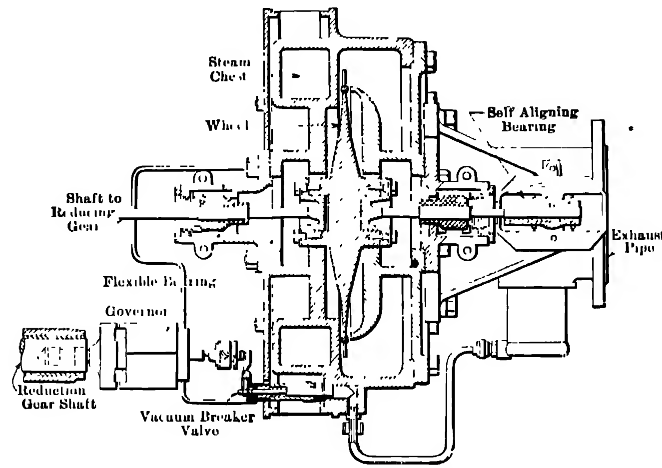


FIG. 277. De Laval "Class A" Steam Turbine.

distributed around the circumference, but, in the later types, are arranged in groups. As illustrated in Fig. 278, the nozzles are placed at an angle of 20 degrees with the plane of the disc. The steam is expanded in the nozzles to the existing back pressure before it impinges at high velocity against the blades. After giving up its energy, the steam passes into the body of the casing and out through the exhaust opening.

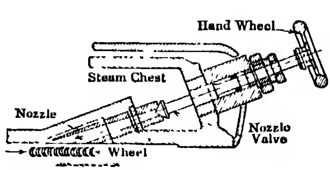


FIG. 278. Nozzle Arrangement, De Laval "Class A" Turbine.

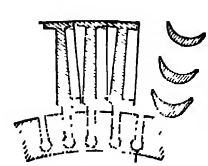


FIG. 279. Turbine Blading, De Laval "Class A" Turbine.

"Class A" De Laval turbines have been built in sizes ranging from 17 to 700 hp. The diameter of the wheel varies from 4 in. in the smallest size to 30 in. in the largest. The speeds vary from 10,600 r.p.m. for the largest size to 30,000 r.p.m. in the smallest, corresponding to the peripheral velocities of 1310 to 520 ft. per sec. respectively. The speeds are reduced by the gearing 10 to 1.

While this particular type of turbine is no longer constructed, the elementary theory involved will be discussed at some length because the same principles apply to all types of impulse turbines and the fewness of parts permits of simple analysis.

The maximum theoretical power developed by a jet of steam flowing through a nozzle is dependent only upon the *weight* of steam flowing per unit of time and the discharge or spouting *velocity*. Therefore, the higher the spouting velocity for a given rate of flow the greater will be the power developed and the higher the efficiency.

The maximum *weight* of steam discharged through a nozzle of any shape and for a given initial pressure is determined by the *area* of the narrowest cross section or *throat*.

To obtain the maximum *velocity* at the exit or *mouth*, for a given rate of flow, the nozzle should be proportioned so that expansion to the external pressure into which the nozzle delivers shall take place within the nozzle itself. If expansion in the nozzle is incomplete, sound waves will be produced and there will be irregular action and loss of energy. On the other hand, if expansion in the nozzle is carried below that of the external pressure at the mouth, sound waves will be produced with subsequent loss of energy even greater than in the former case.

Experimental and mathematical investigations indicate that the pressure at the narrowest section of an orifice or the throat of a nozzle through which steam is flowing falls to approximately 0.58 of the initial absolute pressure (with resultant velocity of about 1400 to 1500 ft. per sec.) and any further fall in pressure must take place beyond the narrowest section. Thus, for back pressures greater than 0.58 of the initial (conveniently taken as 3/5), maximum exit velocity may be obtained from orifices of nozzles of uniform cross section or with sides **convergent**. For back pressure less than 0.58 of the initial, the nozzle must first **converge** from inlet to throat and then **diverge** from throat to mouth in order to obtain maximum velocity. Without the divergent portion of the nozzle, the jet will begin to spread after passing the throat, and its energy will be given up in directions other than that of the original jet.

Figure 280 shows a section through a theoretically proportioned expanding nozzle. The cross section of the tube at any point n may be calculated by means of equation

$$A_n = WS_n \div V_n \tag{168}$$

in which

A_n = area in sq. ft.,
W = maximum weight of steam discharged, lb. per sec.,
S_n = specific volume of the steam at pressure P_n.

For wet steam $S_n = x_n u_n + \sigma$,

in which

x_n = quality of steam at pressure P_n after adiabatic expansion from pressure P_1,

u_n = specific volume of saturated steam at pressure P_n,

σ = volume of 1 lb. of water corresponding to pressure P_n.

This quantity is very small compared with that of the steam, and may be neglected.

$$V_n = 223.7 \sqrt{H_1 - H_n}$$

By substituting H_n = heat content corresponding to pressure $P_n = 0.58P_1$ in equations (157) and (168) the area at the throat may be readily determined. The cross-sectional area for other points in the tube may be determined in a similar manner by assigning values of H_n corresponding to the various pressures.

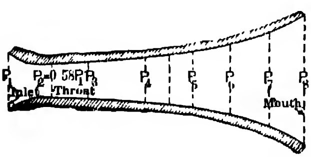


FIG. 280. Theoretically Proportioned Expanding Nozzle.

In case of a perfect nozzle $H_1 - H_n$ represents the heat given up toward producing velocity by adiabatic expansion from pressure P_1 to P_n. In the actual nozzle the frictional resistance of the tube serves to increase its dryness fraction, but in doing so it decreases the amount of energy the steam is capable of giving up towards increasing its own velocity. If y one-hundredths of the heat, $H_1 - H_n$, is utilized in overcoming frictional resistance, then the resulting velocity will be

$$V = 223.7 \sqrt{(1 - y)(H_1 - H_n)}. \qquad (169)$$

The quality of the steam after expanding to P_n against the resistance will be higher by an amount

$$I_n = \text{increase in quality} = y(H_1 - H_n)/r_n \qquad (170)$$

in which

r_n = heat of vaporization at pressure P_n.

The curves in Fig. 281, calculated by means of equations (163) and (168), show the relationship between velocity, quality, pressure, and kinetic energy for all points in a theoretically perfect nozzle expanding 1 lb. of dry steam per sec. from an initial absolute pressure of 190 lb. abs. to a condenser pressure of 1 lb. abs.

The curves in Fig. 282 are based upon the experiments of Gutermuth and show the effect of a few shapes of nozzles and orifices on the actual

weight of steam discharged for various rates of initial and final pressures, the smallest section of the tube remaining constant.

The nozzles of most commercial types of steam turbines are made with straight sides as in Fig. 278, so that only the area at the mouth need be determined in addition to that at the throat in order to lay out the shape of the tube.

Equations (157) and (168) are general and are applicable to steam of any quality, wet, dry, or superheated.

The diameter at the throat may be calculated within an error of 1 to 2 per cent, for the range of pressures usually encountered, by means of **Grashof's formula.**

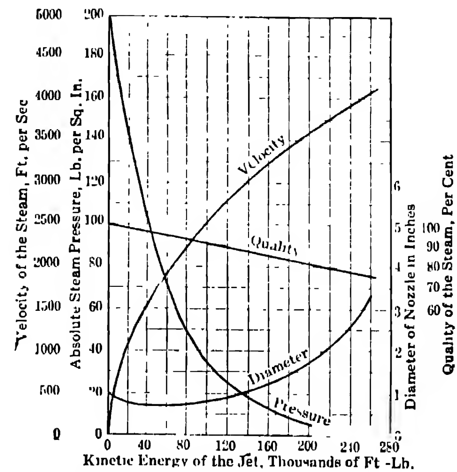


FIG. 281. Characteristics of a Theoretically Proportioned Expanding Nozzle.

For dry or wet steam when $P_n = < 0.58P_1$

$$w' = 60a_oP_1^{0.97}\sqrt{x_1} \tag{171}$$

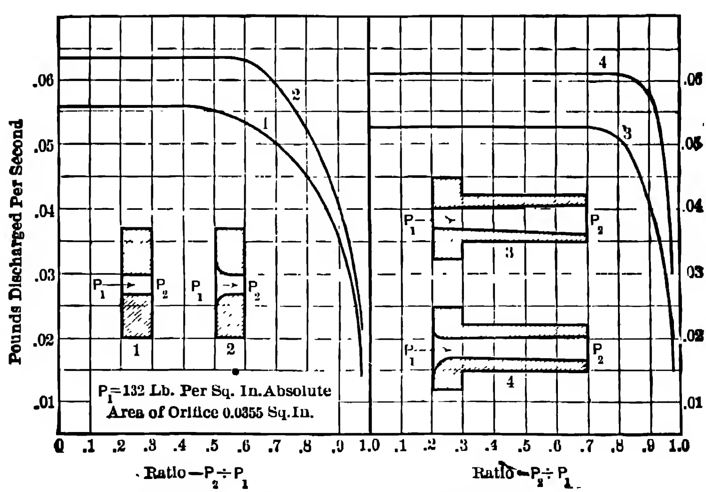


FIG. 282. Flow of Steam through Nozzles.

For superheated steam when $P_n = < 0.58P_1$

$$w' = 60a_oP_1^{0.97} \div (1 + 0.00065t_s), \tag{172}$$

in which

w' = actual weight of steam discharged, lb. per hr.,
a_o = area of the throat, sq. in.,
P_1 = initial absolute pressure, lb. per sq. in.
x_1 = initial quality,
t_s = degree of superheat, deg. fahr.

Goudie (*Steam Turbines*) gives the following rule for superheated steam

$$w' = 18.9\, a_o \sqrt{P_1 \gamma_1} \qquad (173)$$

in which

γ_1 = density of the steam at pressure P_1; other notations as above.

For back pressures higher than the critical or $P_n > 0.58 P_1$ the fundamental equation (157) offers the simplest solution. Approximate results for this condition may be obtained by multiplying equations (171) and (172) by a factor K

$$K = 2.182 \sqrt{c(1 - 1.19c)}, \qquad (173a)$$

in which

$$c = 1 - (P_n \div P_1).$$

When a divergent nozzle having an actual area ratio r (= mouth area ÷ throat area) is used for steam pressure having a ratio R (= mouth area ÷ throat area for pressure ratio P_n/P_i), a percentage nozzle-mouth error of a value $c_1 = 100(r - R) \div r$, which may be positive or negative is introduced. The following table gives the velocity efficiency or ratio of probable actual exit velocity to the theoretical velocity for various nozzle-mouth errors, assuming the correctly proportioned nozzle to have a velocity efficiency of 97 per cent.

Nozzle-mouth error, c_1	−40	−30	−20	−10	0	10	15	20	25	20
Velocity efficiency, per cent	93.5	94 8	95.9	96.7	97	96 7	96.3	95.3	93.6	90 6

When the actual expansion ratio of the nozzle is greater than required, the nozzle is said to be over-expanded; when smaller, under-expanded. From the preceding table it appears that it is preferable to have a nozzle under-expanded than over-expanded.

Moyer ("The Steam Turbine," 4th Edition, p. 44) states that the ratio of the area of a correctly proportioned nozzle at the throat a_o to the area at any point a_n is very nearly proportional to the ratio of the pressure at point a_n to the initial pressure, or

$$\frac{a_o}{a_n} = \frac{P_n}{P_i} \qquad (174)$$

The entrance to the tube is rounded by any convenient curve.

The length of the tube may be roughly approximated by the following formula:

$$L = \sqrt{15a_o} \qquad (175)$$

in which

L = length between the throat and mouth, in inches,
a_o = area at the throat, square inches.

Practice shows that the cross section of a nozzle, whether circular, elliptical, square, or rectangular (the latter with rounded corners), has very little influence on the efficiency, provided the inner surfaces are smooth and the ratio of the area at the throat to that of the mouth is correctly proportioned. The *velocity* efficiency of a properly proportioned nozzle with straight sides is about 95 to 97 per cent, corresponding to an *energy* efficiency of 92 to 94 per cent, so that it is not considered worth while to attempt to follow the more difficult exact curves.

Example 38. — Find the smallest cross section of a frictionless, conical, divergent nozzle for expanding 1 lb. of steam per sec. from an absolute initial pressure of 190 lb. to an absolute back pressure of 2 lb. and find six intermediate cross sections where the pressures will be 70, 30, 14.7, 8, 4, and 2 lb. respectively. Compare the velocity and energy of the jet issuing from this nozzle with those of an actual nozzle in which 10 per cent of the heat energy is lost in friction.

Solution. — From steam and entropy tables we find the values of H, x, u, for absolute pressures corresponding to 190, 0.58 × 190 = 110, 70, 30, etc., lb. per sq. in. as follows (theoretical nozzle):

	H	x	u	$S = xu$
P_1 = 190	1197.3	1.00	2.406	2.406
P_2 = 110*	1152.6	0.960	4.047	3.885
P_3 = 70	1117.9	0.932	6.199	5.775
P_4 = 30	1057.2	0.887	13.75	12.27
P_5 = 14.7	1011.3	0.857	26.78	22.95
P_6 = 8	947.8	0.834	47.26	39.29
P_7 = 4	935.6	0.810	90.4	73.2
P_8 = 2	899.3	0.788	173.1	137.0

* $P_2 = 0.58\,P_1$ (= pressure at throat)

If entropy tables or charts are not available, values H_1 to H_8 and x_1 to x_8 must be calculated. (See paragraph 392.)

The different quantities for the theoretical nozzle will be calculated for the exit pressure $P_n = P_8 = 2$ lb. per sq. in. abs.

$$V_8 = 223.7\sqrt{H_1 - H_8}$$
$$= 223.7\sqrt{1197.3 - 899.3} = 3865 \text{ ft. per sec.}$$

$$E_8 = 778\ (H_1 - H_8)$$
$$= 778\ (1197.3 - 899.3) = 232{,}000 \text{ ft-lb.}$$
$$A_8 = WS/V = 1 \times 137/3865 = 0.0353 \text{ sq. ft.}$$
$$d_8 = \sqrt{A(144 \times 4)/\pi} = 13.56\sqrt{0.0353} = 2.54 \text{ in.}$$
$$F_8 = WV_8/g = 3865/32.2 = 120 \text{ lb.}$$

THEORETICAL NOZZLE

Quantity....		V Ft. per Sec	E Ft-lb	A Sq. Ft	d In	F Lb.
Formula.		(163)	(153)	(168)		(156)
Pressures	110	1496	34,767	00259	0 693	46 4
	70	1995	61,853	.00269	0 702	62 0
	30	2650	107,485	.00461	0.919	82.3
	14 7	3053	144,742	00745	1.1	94.8
	8	3339	173,207	0119	1.46	103.7
	4	3624	203,968	.0202	1.92	112.5
	2	3865	232,000	.0353	2 54	120 0

In the actual nozzle these values will be modified because of the frictional losses. Thus, for $P_n = 2$ lb.,

$$V_8 = 223.7\sqrt{(1 - y)\ (H_1 - H_8)}$$
$$= 223.7\sqrt{(1 - 0.1)\ (1197.3 - 899.3)} = 3667 \text{ ft. per sec.}$$
$$E_8 = 778\ (1 - 0.1)\ (1197.3 - 899.3) = 208{,}800 \text{ ft-lb.}$$
$$x_8' = x_8 + I_8 = x_8 + y(H_1 - H_8)/r_8$$
$$= 0.788 + 0.1\ (1197.3 - 899.3)/1021$$
$$= 0.788 + 0.029 = 0.817.$$
$$A_8 = Wx_8'u_8/V_8 = 0.817 \times 173.1/3667 = 0.0386 \text{ sq. ft.}$$

from which

$$d_8 = 2.66 \text{ in.}$$
$$F = WV_8/g = 3668/32.2 = 114 \text{ lb.}$$

These various factors for all given pressures have been calculated in a similar manner and are as follows:

ACTUAL NOZZLE

Quantities........ ...		V Ft. per Sec	E Ft-lb.	$x°$	A Sq. Ft.	d In.	F Ft-lb.
Pressures	110	1420	31,317	.9658	00275	0 711	44.1
	70	1893	55,632	.9414	00286	0 723	58.8
	30	2515	98,257	.9026	.00493	0.951	78.1
	14.7	2894	130,050	.876	0080	1.2	98.8
	8	3168	155,858	.856	.0127	1.53	98.4
	4	3438	183,581	836	0220	2.01	106.8
	2	3667	208,800	817	.0386	2.66	114.0

Many of these values may be determined directly from the *Mollier* or total heat-entropy diagram as described in paragraph 386; in fact, the Mollier diagram has to all intents and purposes supplanted the steam tables in this connection. For superheated steam the diagram is extremely useful in avoiding laborious calculations.

Figure 283 gives a diagrammatic arrangement of the blades in a single-stage De Laval turbine. The nozzle directs the steam against the blades with *absolute* velocity V_1 and at an angle α with the plane of the wheel XX. Since the wheel is moving at a velocity of u ft. per sec., the velocity v_1 of the steam *relative* to the wheel is the resultant of V_1 and u. The angle β_1 between v_1 and XX will be the proper blade angle at entrance. If the

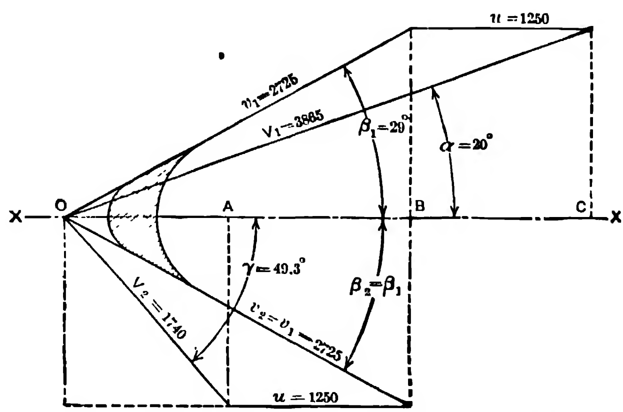


FIG. 283. Velocity Diagram. Ideal Single-pressure, Single-velocity Stage Turbine.

blade curve makes this angle with the direction of motion of the wheel, no shock will be experienced when the steam enters the blades. For convenience in construction the exit angle β_2 is made the same as the entrance angle β_1. Neglecting frictional losses in the blade channels, the *relative* exit velocity will be $v_2 = v_1$, and the resultant of v_2 and u is the *absolute* velocity, V_2. The impulse exerted by the jet in striking the vanes is Wv_1/g, and its component in the direction of motion is $Wv_1 \cos \beta_1/g = W(V_1 \cos \alpha - u)/g$. As the jet leaves the vanes the impulse is $-Wv_2 \cos \beta_2/g = -W(V_2 \cos \gamma + u)/g$.

The *total force* acting on the vanes, or the actual driving impulse, is

$$P = W/g \times \{ V_1 \cos \alpha - u - [- (V_2 \cos \gamma + u)] \} \qquad (176)$$

$$= W/g \times (V_1 \cos \alpha + V_2 \cos \gamma). \qquad (177)$$

Equation (176) may also be expressed

$$P = W/g \times 2(V_1 \cos \alpha - u). \qquad (177a)$$

The resultant *axial force* or *end thrust* is

$$F = W/g \times (V_1 \sin \alpha - V_2 \sin \gamma). \tag{178}$$

Evidently if $\alpha = \gamma$ and $V_1 = V_2$ there will be no end thrust, since $V_1 \sin \alpha - V_2 \sin \gamma$ will be zero.

The *work done* is

$$Pu = Wu\ (V_1 \cos \alpha + V_2 \cos \gamma)/g \tag{179}$$

or, using equation (177a) in place of (176)

$$\begin{aligned} Pu &= W/g \times 2u\ (V_1 \cos \alpha - u) \\ &= W/g \times 2\ (uV_1 \cos \alpha - u^2). \end{aligned} \tag{180}$$

By making the first derivative equal to zero

$$\frac{d}{du}\left[\frac{W}{g} 2(uV_1 \cos \alpha - u^2)\right] = V_1 \cos \alpha - 2u = 0,$$

or

$$u = \tfrac{1}{2} V_1 \cos \alpha \tag{181}$$

That is, for *any nozzle angle* α the work done, Pu, has its greatest value when $u = \frac{1}{2} V_1 \cos \alpha$, or $\gamma = 90$ degrees, whence

$$Pu = W/2g \times V_1^2 \cos \alpha \tag{182}$$

The work for *any initial velocity* V_1 becomes a maximum when $\alpha = 0$ and $u = \frac{1}{2}V_1$. *This condition can only occur for a complete reversal of jet and zero final velocity.* Substituting $\alpha = 0$ and $u = \frac{1}{2}V_1$ in equation (181) and reducing, we have

$$Pu = E_4 = WV_1^2 \div 2g$$

which is necessarily the same as equation (157).

In the actual turbine the various velocities will be less than those so obtained, on account of the frictional resistance in the blades, and the velocity diagram should be modified accordingly.

Example 39. — Lay out the blades (theoretical and actual) for the nozzle in the preceding example, assuming that the jet impinges against the wheel at an angle of 20 degrees and that the peripheral velocity is 1250 ft. per sec. Weight of steam flowing, 1 lb. per sec.

Solution. — *Theoretical Case.* Lay off $V_1 = 3865$ ft. per sec. in direction and amount as shown in Fig. 283 and combine it with $u = 1250$ ft. per sec.; this gives v_1, the relative entrance velocity, as 2725 ft. per sec., and β, the entrance angle, as 29 degrees.

Lay off $v_2 = v_1$ at an angle $\beta_2 = \beta_1$ and combine with u; this gives V_2, the *absolute* exit velocity, as 1740 ft. per sec.

The theoretical energy available for doing work is

$$E = W/2g \times (V_1^2 - V_2^2) = 1/64.4 \times (3865^2 - 1740^2) = 185,000 \text{ ft-lb.}$$

The difference between 232,000 and 185,000 = 47,000 ft-lb. is evidently the kinetic energy lost in the exhaust due to the exit velocity.

The pressure exerted by the steam on the buckets is

$$P = W/g \times (V_1 \cos\alpha + V_2 \cos\gamma)$$
$$= 1/32.2 \times (3865 \times 0.9397 + 1740 \times 0.65166) = 148 \text{ lb.}$$

The theoretical impulse efficiency is

$$(V_1^2 - V_2^2)/V_1^2 = (\overline{3865}^2 - \overline{1740}^2)/\overline{3865}^2 = 0.797.$$

The theoretical hp. developed is

$$\text{Hp.} = 185,000/550 = 336.$$

Theoretical steam consumption per hp-hr. is

$$3600/336 = 10.7 \text{ lb.}$$

Actual Case. — Proceed as in the theoretical case, using the actual absolute velocity $V_1 = 3865\sqrt{1-y} = 3865\sqrt{1-0.10} = 3667$ ft. per sec. in place of the theoretical value $V_1 = 3865$. Lay off $V_1 = 3667$ at an angle of 20 degrees as before and combine with $u = 1250$, Fig. 284.

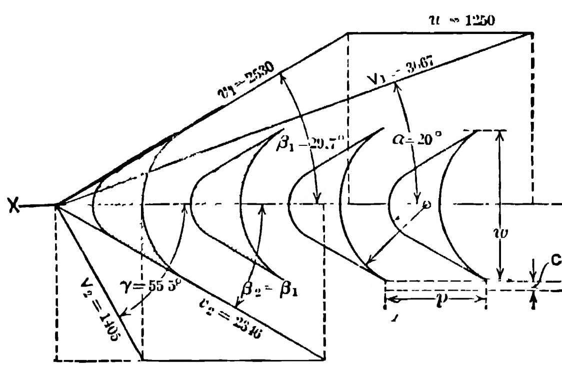


Fig. 284. Velocity Diagram as Modified by Friction Losses.

The resultant $v_1 = 2530$ is the velocity of the jet relative to the wheel, and the entrance angle β is found to be 29.7 degrees. The relative exit velocity v_2 will be less than v_1 because of the blade friction.

Assume the loss of energy ϕ between inlet and exit of the blades to be 14 per cent; then, since the velocity varies as the square root of the energy,

$$v_2 = v_1\sqrt{1-\phi} \tag{183}$$
$$= 2530\sqrt{1-0.14} = 2346 \text{ ft. per sec.}$$

The resulting absolute velocity V_2 is found from the diagram to be $V_2 = 1405$ ft. per sec.

Since the loss of energy in the nozzle is

$$V_1^2 - (1 - y)\ V_1^2 \div 2g, \tag{184}$$

and that in the blade

$$v_1^2 - (1 - \phi)\ (v_1^2 \div 2g), \tag{185}$$

the remaining energy, deducting both losses in the nozzle and the blades, is

$$W/2g \times (V_1^2 - yV_1^2 - \phi\, v_1^2 - V_2^2) \tag{186}$$

$$= 1/64.4 \times (3865^2 - 0.1 \times 3865^2 - 0.14 \times 2530^2 - 1405^2)$$
$$= 164{,}200, \text{ ft-lb.}$$

The losses due to windage, leakage past the buckets, and mechanical friction must be deducted from these figures to give the actual energy available for doing useful work. Assuming a loss of 15 per cent due to this cause, the work delivered is

$$0.85 \times 164{,}200 = 139{,}570 \text{ ft-lb.}$$

The efficiency in the ideal case was found to be 0.797 and the available energy 185,000 ft-lb.

The efficiency, deducting the loss due to friction, etc., is

$139{,}570 \times 0.797 \div 185{,}000 = 0.60.$
Hp $= 139{,}570/550 = 254.$

Steam consumption per hp-hr. is

$3600/254 = 14.2$ lb.

The heat consumption, B.t.u. per hp. per min. is

$14.2(1197.3 - 94)/60 = 260.$

Assuming the r.p.m. to be 10,000, the mean diameter of the wheel to give a peripheral velocity of 1250 ft. per sec. is

$1250 \times 60 \div 10{,}000 \times 3.14 = 2.39$ ft., or 29.6 in.

The determination of the height and width of vanes, clearance between nozzles and blades, etc., are beyond the scope of this work, and the reader is referred to the accompanying bibliography.

The ratio of exit to inlet velocity is called the blade or bucket velocity coefficient. The following table gives the values of this coefficient for the usual shape of impulse turbine blades. The values include all losses between the nozzle mouth and entrance to the exhaust opening. (Marks' Mechanical Engineers' Handbook, p. 984.)

Velocity relative to blades, ft. per sec.	200	400	600	800	1000	1500	2000	2500	3000	4000
Blade velocity coefficient.......	0 953	0.918	0.888	0.863	0.841	0.801	0.774	0.754	0.739	0.716

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Reaction Vanes with Radial and Axial Clearance, Feb. 26, 1924, p. 329.

Comparison of Radial- and Axial-flow Turbines. Power, Jan. 8, 1924, p. 50.

204. Single-pressure, Compound-velocity-stage Impulse Turbine. — In this type of impulse turbine, the steam is expanded down to the existing back pressure in a single set of nozzles just as in the single-velocity stage machine, but no attempt is made to absorb the kinetic energy of the jet in a single passage through the vanes or buckets on the wheel by maintaining a high peripheral velocity. Instead, the blade velocity is fixed at some point much lower than half the effective velocity of the jet, so that steam leaves the buckets with considerable residual energy. In order to operate with low blade velocity and at the same time utilize part of the residual energy, the steam leaving the wheel may be guided by a set of stationary reversing vanes to another wheel. If the steam leaves the second wheel at a high velocity, a third set of reversing and moving vanes may be employed. This procedure may be continued for any number of stages or until the steam leaves the last row of moving vanes at practically zero velocity. The same result may be obtained by redirecting the steam from the reversing buckets upon the same wheel which receives the initial impulse of the jet. For maximum theoretical efficiency, the number of velocity stages necessary for a given blade velocity is equal to the initial jet or spouting velocity divided by twice the blade velocity. Thus, for an initial jet velocity of 4000 ft. per sec., there should be 2, 4 and 8 stages for blade velocities of 1000, 500 and 250 ft. per sec. respectively. In the actual turbine these values would be modified because of the frictional resistances in the nozzles and blades, windage, leakage and the angles of the vanes.

The single-stage **Curtis**, **"Class C" De Laval**, single-stage **Moore**, **"Type K" Kerr**, non-condensing **Terry**, **Westinghouse Impulse**, and **Sturtevant** are well-known examples of this class of steam turbine.

Figure 285 shows a section through a De Laval "Class C" single-pressure, two-row, velocity-stage, non-condensing turbine illustrating the well-known "Curtis" principle. The rotor consists of a single forged-steel disc, fitted at the periphery with two rows of bronze, monel metal, or forged-steel blades, the material depending upon the initial conditions

of the steam. The blades are similar in design to those of the single-stage geared type. The nozzles are of the diverging type and are machined in a removable nozzle plate bolted to the side of the wheel case. The guide vanes are of the same design as those on the rotor and are fastened in a similar manner to removable steel segments. These segments are bolted to the nozzle plate in such a manner that the guide vanes are intermediate between the two rows of moving vanes. The governor is of the centrifugal type, mounted on the main turbine shaft and operating a double-seated balanced valve. Machines suitable for high speeds, such as small,

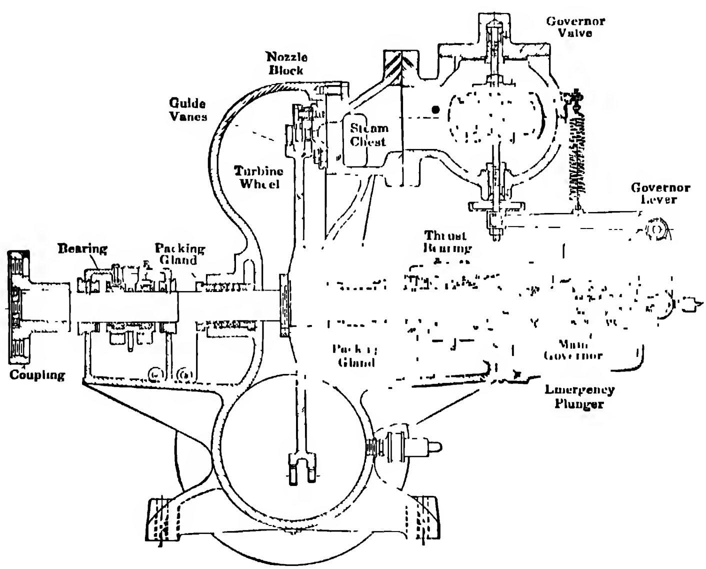


FIG. 285. De Laval "Class C" Steam Turbine.

high-head centrifugal pumps, centrifugal blowers and compressors, and small direct-current generators, can be directly connected to the turbine through the coupling, but for lower-speed machines it is necessary to use a reduction gear between the turbine and the driven machine. The two-row wheel turbine is intended for small powers only. In the larger machines there are several wheels and intermediates depending upon the power requirements, the initial and final steam conditions, and the desired peripheral velocity. The governors for the larger machines are either of the Jahns type, mounted on a vertical spindle and driven by the turbine shaft through worm gearing, or of the hydraulic relay type. All sizes are equipped with a safety stop and quick-operating trip valve. "Class C" turbines are available in various sizes up to 1200 hp.

The **Kerr velocity-stage** turbine differs from the "Class C" De Laval in that there is but a single row of vanes on each disc. The present construction consists of two or three single-row wheels, and is intended primarily for driving power plant auxiliaries in sizes from 5 to 600 hp. without reduction gearing.

The velocity diagrams may be constructed in a manner similar to that of any single-pressure stage in the Curtis turbine.

Figure 286 gives a general view of a **Terry non-condensing** turbine illustrating the single-pressure compound-velocity-stage principle as applied to a single wheel of the re-entrant type. Steam is expanded down to the existing back pressure in one or more nozzles, depending upon the size of the turbine. The resulting high-velocity jet enters the side of the wheel bucket and its direction is reversed 180 degrees. As this single reversal uses but a portion of the available energy, the steam is caught in a reversing chamber and returned again to the wheel. This process is repeated several times. As the wheel and buckets are cut from a single piece of steel, there are no parts to become loose or worn out. The reversing chambers or buckets are made of special alloy metal and are arranged in groups on the inner surface of the turbine casing, each group being supplied with a separate nozzle. The governor is of the fly-ball throttling type and is mounted on one end of the shaft. In the larger sizes having more than one nozzle, the load may also be manually controlled by opening and closing the nozzles. Terry non-condensing turbines are made in a number of sizes ranging from 5 to 600 hp. and may be direct connected or geared, depending upon the speed requirements of the driven machines.

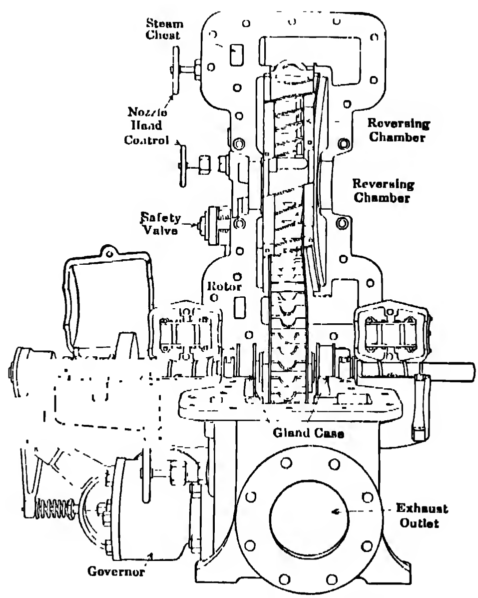


FIG. 286. Terry Non-condensing Steam Turbine.

The **Westinghouse impulse** turbine is of the single-pressure compound-velocity type employing a single wheel with one row of vanes and a section or block of stationary vanes for redirecting the steam onto the rotor.

While the arrangement bears a resemblance to the Terry turbine, the steam passes but twice through the blades on the rotor and the action is equivalent to that of a machine having two rows of rotating blades, and

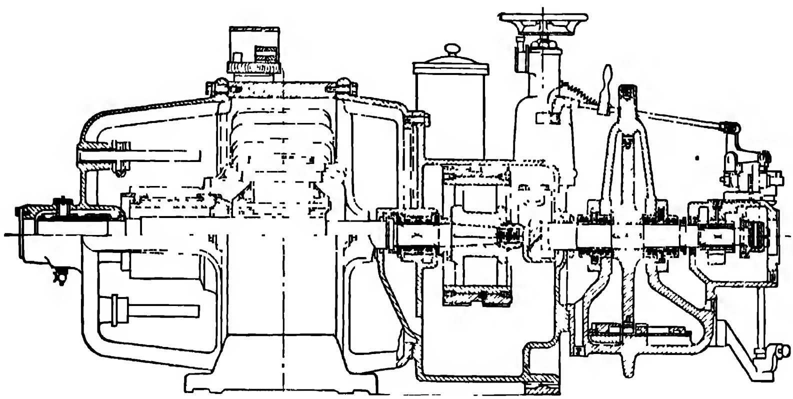


FIG. 287. Westinghouse Impulse Turbo-Generator — Geared Set.

the best economy is obtained when the ratio of blade velocity to that of the jet is 0.23. These turbines are constructed in sizes ranging from a fraction to 3000 hp. For very high-speed service the turbine is direct connected to the driven machinery, but for lower-speed drives a reduction gear is used. Figure 287 shows the general assembly of a geared turbo-generator set and Fig. 288 that of a fixed-reduction gear of the type used for small powers at comparatively low speed. When the pinions are small in diameter and the width of the face is narrow, a slight misalignment of the bearings does not materially affect the tooth contact. For heavier loads the pinion diameter and the width of the face must be increased, but the permissible limit for pitch-line speed is soon reached and further increase in power-transmission requirements must be provided for by an increase in width of face without an increase in diameter. When the total face is more than three diameters, it is the practice of the Westinghouse Company to

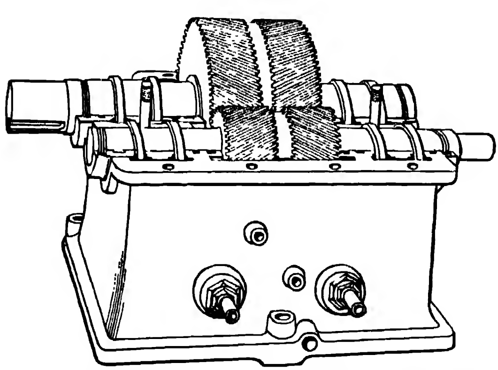


FIG. 288. Westinghouse Reduction Gearing for Small Units.

carry the pinion in bearings attached to a frame within the housing and flexibly supported from the housing through a short section of steel "I" beam. The flexibility of the web of the "I" beam section will permit a uniform distribution of the pressure throughout the width of the gears. While Westinghouse impulse turbines are equipped with automatic stop governors which close the throttle if the speed is 10 per cent above normal, additional precaution is taken to prevent rupture of the shaft in case the automatic stop fails to function. This is accomplished by surrounding the rotor hubs with massive steel rings which will confine the deflection of the shaft within narrow limits and act as a brake. Heavy ribs surrounding the periphery of the rotor accomplish the same purpose on the smaller sizes of machines.

Valve Gears on Small Westinghouse Direct-driven Turbines: Power, Nov. 6, 1923, p. 730.

205. Compound-pressure, Single-velocity-stage Impulse Turbines (Rateau Type). — It has been shown that the velocity of the jet issuing from a nozzle varies with the square root of the heat drop. If the entire expansion from throttle valve to exhaust opening takes place in a single set of nozzles, the resultant jet velocity is very high, and in order to realize maximum efficiency the peripheral velocity must be approximately half that of the jet or else the velocity must be compounded as described in the preceding paragraph. If, instead of expanding completely in one set of nozzles, the heat drop is effected step by step through a series of nozzle sets, the jet velocity will be reduced by an amount equal to the square root of the number of nozzle sets. For example, if H_1 and H_n represent, respectively, the initial and final heat content of the steam, the jet velocity for complete expansion in one set of nozzles is $V = 224 \sqrt{H_1 - H_n}$. If the heat drop, $H_1 - H_n$, is equally divided among n sets of nozzles the heat drop per stage will be $(H_1 - H_n)/n$ and the jet velocity will be

$$V = 224 \sqrt{(H_1 - H_n)/n}.$$

Thus a four-pressure-stage machine with single velocity stages can operate efficiently at one half the speed of a single-pressure-stage machine; a 16-stage at one-fourth speed; a 64-stage at one-eighth speed and so on. Attention should be called to the fact that in simple velocity compounding the entire turbine casing, with the exception of the steam chest, is under a pressure corresponding to that of the back pressure, whereas in pressure compounding each stage is under a pressure increasing in amount from the last to the first stage. The Ridgway is the best-known American turbine built on the straight Rateau basis.

Figure 289 shows a section through a ten-stage **Ridgway** turbine illustrating an application of the straight Rateau principle. The rotor consists of a series of steel discs keyed and shrunk on a rigid shaft and separated from each other by steel collars. A series of buckets machined from solid bars of a special alloy are secured to the periphery. The smaller buckets are fastened to the wheels by rivets through their shanks, and the larger ones are driven into slots and peened. The diaphragms containing the nozzles are secured to the casing and arranged so as to form a separate compartment or cell for each wheel. The casing and diaphragms are split horizontally so that the machine can be readily opened for in-

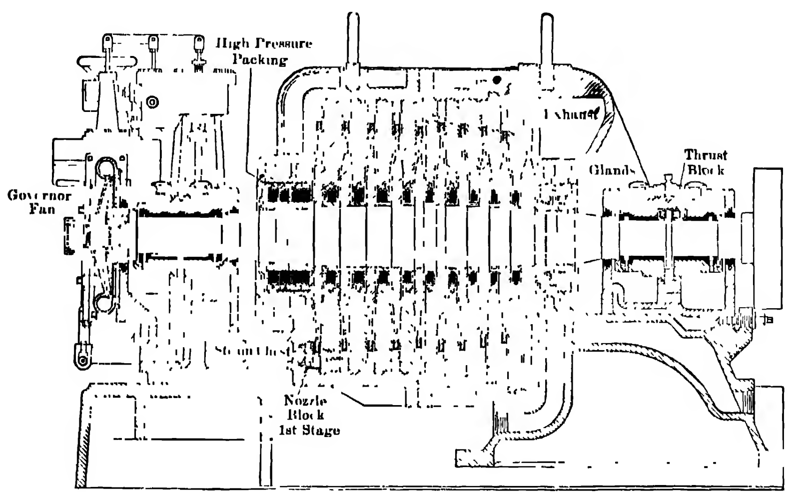


FIG. 289. Ridgway-Rateau High-pressure Steam Turbine.

spection. Nozzles of small area are solid castings of special alloy machined and bolted to the diaphragm. In the later stages, where the nozzle areas are large and extend all the way round the periphery, the blades forming the nozzle are cast in place in the diaphragm. The operation of the turbine is as follows: Steam enters the turbine through the governor valve to the steam chest in which is located the first set of nozzles. Partial expansion takes place through the first set of nozzles and the kinetic energy is imparted to the first wheel through the medium of the buckets or vanes. Steam is discharged from these buckets at a very low velocity and is again partially expanded through the second set of nozzles in the second diaphragm. The resulting kinetic energy is absorbed by the buckets on the second wheel. This process is repeated in each stage. The arrangement of vanes and nozzles is shown in Fig. 290. It will be noted that the nozzle areas and the areas between the vanes gradually

increase from the first to the last stage to allow for the increased volume of steam as effected by reduction in pressure. The governor is of the pneumatic type and its operation is as follows: Air pressure, generated by a centrifugal fan, attached to the end of the turbine shaft is transmitted to two light pistons the movement of which is resisted by a spring. The upper end of the stem carrying these pistons is pivoted to one end of a floating lever. Movement of the lever is transmitted to a small pilot valve, which in turn admits oil pressure to the piston attached to the main throttle valve. The movement of the main valve stem returns the pilot valve to its central position. This equalizes the pressure on the top and bottom of the main piston and arrests its movement, thereby maintaining a fixed throttle opening for a given speed.

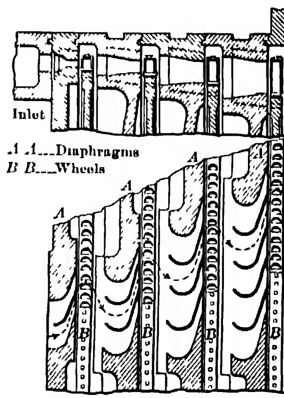


FIG. 290. Arrangement of Vanes and Nozzles, Ridgway-Rateau Turbine.

An eight-stage, compound-pressure, single velocity-stage turbine operating non-condensing at 190 lb. initial absolute pressure would show about the following conditions: (All friction and leakage losses neglected and final velocity in each stage assumed to be zero.)

$$H_1 = 1197.3 \text{ B.t.u. per lb.},$$
$$H_n = 1012.5 \text{ B.t.u. per lb.},$$
$$\text{Total heat drop} = H_1 - H_n = 1197.3 - 1012.5 = 184.8.$$
$$\text{Heat drop per stage} = 184.8 \div 8 = 23.1.$$
$$\text{Stage velocity} = 224\sqrt{23.1} = 1080 \text{ ft. per sec.}$$

Stage	Heat Content	Pressure, Lb Abs.	Quality, Per Cent	Specific Volume Cu. Ft. per Lb.
Admission	1197 3	190	100	2 41
1	1174 2	145	97 9	3 04
2	1151 1	109	95 9	3 93
3	1128 0	80	94.0	5.14
4	1104 9	58	92 2	6 77
5	1081.8	42	89 6	8 96
6	1058 7	30	88.8	12.07
7	1035 6	21	87 3	16.33
8	1012 5	Atmospheric	85.8	22 55

The heat content at the end of expansion in each stage is obtained by subtracting 23.1 B.t.u. from the heat content of the preceding stage. The

corresponding pressures, quality, and specific volume may be calculated as shown in Chapter XXII or they may be taken directly from the Mollier diagram and similar graphical charts.

In the actual turbine, only 50 to 75 per cent of the heat theoretically available is transformed into useful work. A small portion is lost by gland leakage, radiation, and bearing friction, and the balance is retransformed from kinetic energy into potential energy by eddying, fluid friction, and blade leakage. The efficiency of each stage is less than that of the turbine as a whole, since the increase in heat content due to friction, etc., is available for transformation into useful work in the succeeding stages. To find the actual pressure condition in each stage, allowing for the various losses, it is necessary to correct the theoretical quantities for these losses. See "Energy and Pressure Drop in Compound Steam Turbines," by F. E. Cardullo, *Proc. A.S.M.E.*, Feb., 1911, and paper read by Professor C. H. Peabody, *Proc. Society of Naval Architects and Marine Engineers*, June, 1909. Consult also, "The Steam Turbine Expansion Line on the Mollier Diagram and a Short Method of Finding the Reheat Factor," by Edgar Buckingham, Bul. No. 167, 1911, U. S. Bureau of Standards.

206. Compound Pressure, Compound Velocity-stage Impulse Turbines. — It has been shown that the bucket speed for a given heat drop

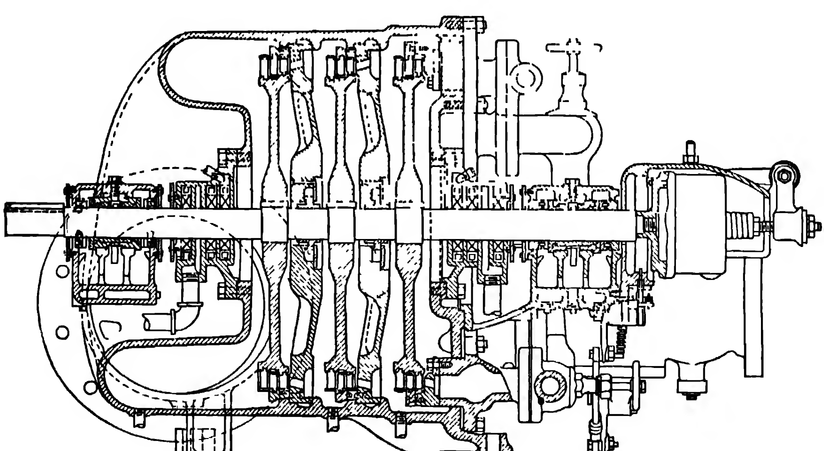


Fig. 291. Assembly of Three-stage Curtis Turbine for Mechanical Drive.

may be reduced, without lowering the economy, by compounding the velocity or the pressure. In case of pure velocity compounding, the pressure throughout each stage is constant, but the velocity of the steam

across the blades is very high in the first stages and gradually decreases from steam inlet to exhaust outlet. In pressure compounding, the stage velocity is practically constant, but the steam pressure is gradually decreased from inlet to outlet. Moreover, for a given heat drop there will be n velocity stages as against n^2 pressure stages. To take advantage of the low spouting velocity of the pressure type and at the same time decrease the number of stages, both pressure and velocity may be compounded in the same turbine. Various combinations of pressure and velocity compounding are to be found in the commercial machine.

The **Curtis** is the best known and the most widely distributed impulse steam turbine in this country. With the exception of the very small and the very large units, Curtis turbines are of the compound-pressure, compound velocity-stage type. The number of pressure stages varies with the size and the service for which the turbine is intended and ranges from one, in a small 10-hp. non-condensing unit, to 23 or more in a 50,000 kw. high-pressure condensing turbo-alternator. Ordinarily there are two velocity stages in the first pressure stage and one velocity stage for each pressure stage throughout the rest of the expansion zone. In the small turbine designed for mechanical drive, there are two velocity stages for each pressure stage and in the large turbo-alternators of 20,000 kw. and over, there is only one velocity stage for each pressure stage. The latter in reality should be classified under the "Compound-pressure, Single-velocity-stage" heading. All Curtis turbines are of the axial-flow type with a horizontally split casing, so that the upper half may be readily taken off for inspection or for removal of the shaft and wheels. The small units operate at speeds ranging from 1200 to 5000 r.p.m. and lower speeds are obtained by the use of reduction gears. Sixty-cycle units from 500 to 9000 kw. rated capacity operate at 3600 r.p.m.; 10,000 to 30,000 kw. units at 1800 r.p.m. and single-cylinder units above 30,000 kw. at 1200 r.p.m.

Figure 292 shows a diagrammatic arrangement of the nozzles and blades in a **three-stage Curtis** turbine. The action of the steam is as follows: Entering at A from the steam pipe, it passes through one or more admission valves B into the bowls C. The number of admission valves depends on the load and their action is controlled by the governor. From bowls C the steam expands through nozzles D and impinges against the first row of moving blades and gives up part of its energy. The steam flowing from the first row of moving blades is reversed in direction by the adjacent stationary vanes and is redirected against the second set of moving blades where it gives up its remaining kinetic energy. From this stage the steam flows at reduced pressure through the nozzles, of

the second stage, which are sufficient in number and size to afford the greater area required by increased volume. In expanding in these nozzles it acquires new velocity and gives up energy to the moving

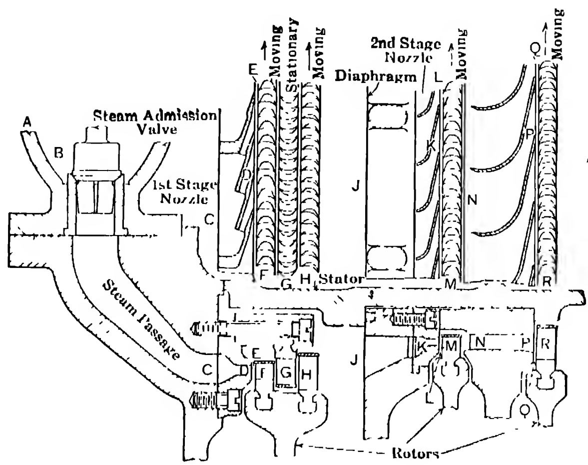


FIG. 292. Diagrammatic Arrangement of Moving and Stationary Elements of Curtis Turbines.

blades as before. This process is repeated through several additional stages.

The rotor consists of 1 to 23 or more steel discs mounted side by side on a horizontal shaft. In some of the earlier designs, the shaft was mounted vertically but this construction has been discontinued. Buckets or vanes of nickel steel, monel metal or nickel bronze according to the condition of the steam, are secured to the periphery by a dovetail-shaped root which fits snugly in a channel of the same section machined in the rim.

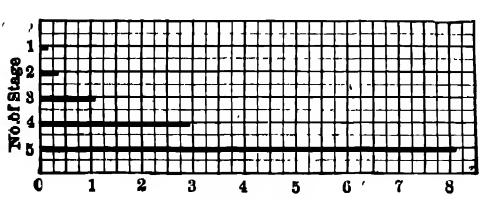


FIG. 293. Steam-belt Area in Five-stage Curtis Turbine.

The tips of the vanes are tenoned and riveted into a shroud ring. The stationary reversing vanes are secured to the casing as illustrated in Fig. 292. Between the revolving wheels is a stationary steam-tight diaphragm which contains the nozzles through which the steam is expanded from the preceding stage. In the older designs, forged-steel nozzles for all stages were cast into the steel diaphragms. In the modern 23-stage 20,000-kw.

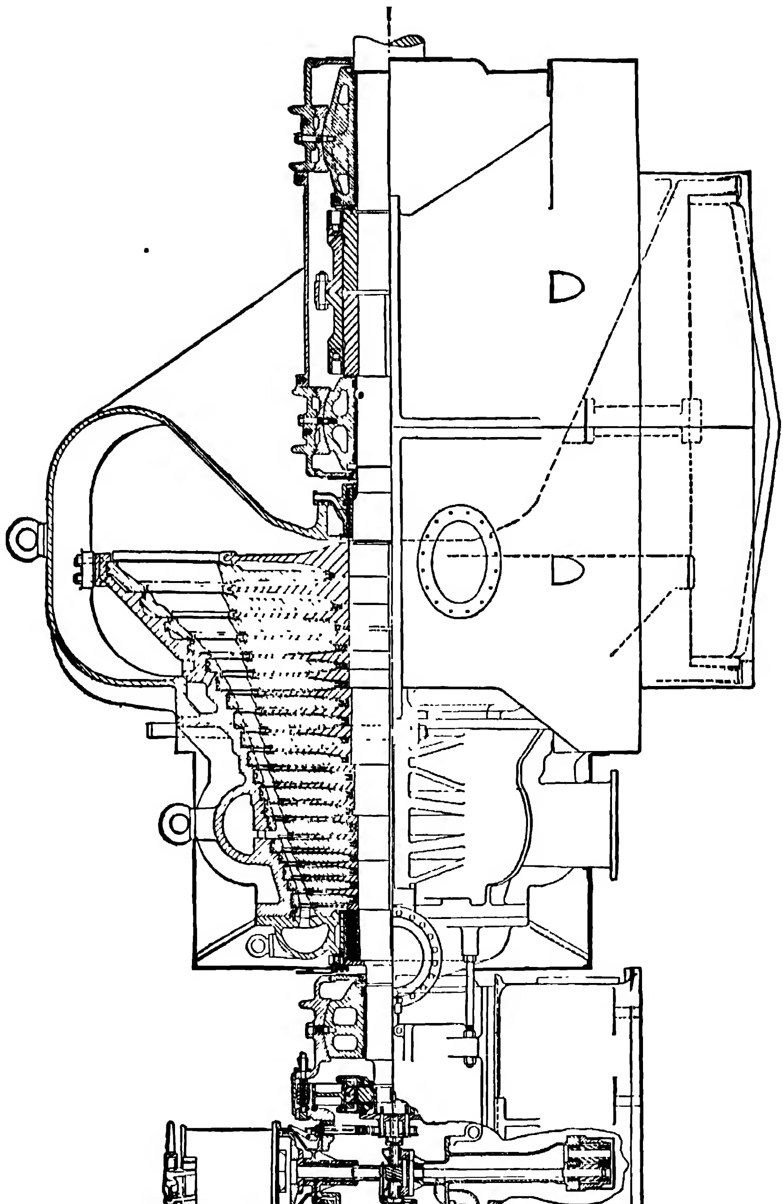


FIG. 294. Assembly of 30,000 kw., 1500 R.p.m., 20-stage Curtis Turbine. (Moyer's "Steam Turbines")

machine, the nozzles in the first 17 stages are of the renewable type and the remaining stages of the older cast construction. In the " renewable " design the supporting spacers, blades and shrouding are of steel, milled and ground, and are assembled in lengths of about 8 to 10 in. The various elements in each section are welded into a rigid construction by copper wire placed alongside the joints and fused in an electric oven. The finished sections of blading are slipped into dovetails in the diaphragms and secured in place. It will be seen from Fig. 293 that vanes and nozzles increase in size in succeeding stages as the pressure falls and the volume increases. The parts are so proportioned that the steam gives up approximately $1/n$ of its energy in each pressure stage, n representing the number of stages. The number of stages and the number of vanes in each stage are governed by the degree of expansion, the peripheral velocity which is practical or desirable, and by various conditions of mechanical expediency. The nozzles extend around a relatively short arc in the periphery of the first stage and increase progressively in number until they extend around the entire wheel in the last stage.

In the smaller machines the speed is controlled by a centrifugal governor mounted on the end of the main shaft and attached through suitable linkage to a single-balanced throttle-valve. The governor actuates a throttling valve of the balanced poppet-valve type. The larger sizes are controlled by an indirect or relay system of the hydraulic type.

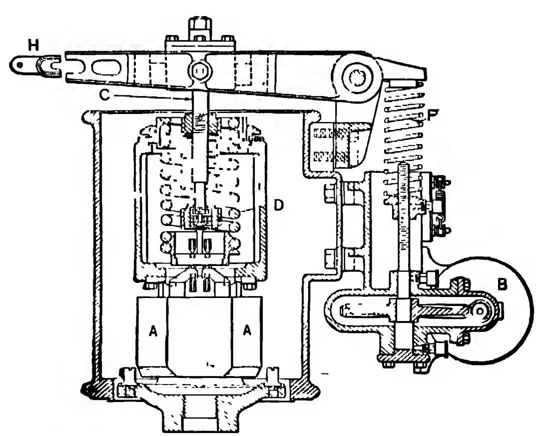


FIG. 295. Main Operating Governor.

Figure 295 gives the general details of the main governor, Fig. 296 a section through the hydraulic cylinder and pilot valve, and Fig. 297 through one of the admission valves of this relay system. Referring to Fig. 295, speed regulation is accomplished by the balance maintained between the centrifugal force of moving weights AA and the static force exerted by spring D. The governor is provided with an auxiliary spring F for varying its speed when synchronizing, the tension of which is varied by a small pilot motor controlled from the switchboard.

The movement of the governor weight is transmitted through rod D to arm H and by means of the latter to floating lever L, Fig. 296.

This floating lever is pivoted on a clamp attached to the pilot valve stem S. The other end of the lever is connected by links to the piston rod R of the operating cylinder. A movement of governor arm displaces the small pistons of the pilot valve from their normal location in which they close the ports of the cylinder. This displacement causes oil to be admitted to the cylinder and the pressure of the oil operates the main piston. The piston rod opens and closes the controlling valves through

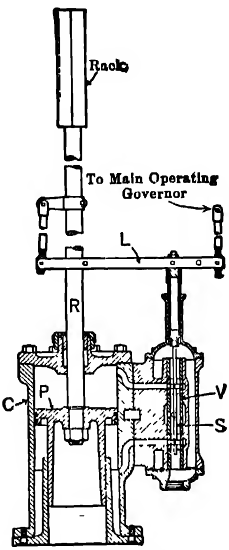


FIG. 296. Assembly of Hydraulic Cylinder.

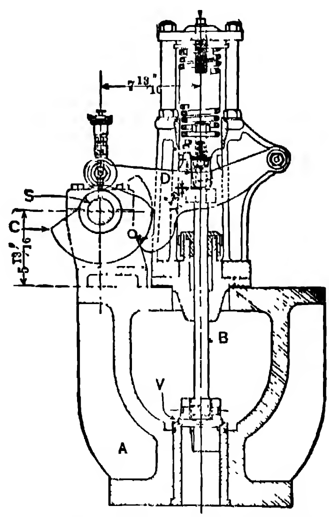


FIG. 297. Admission Valve Control.

the agency of the cam shaft, and at the same time transmits its motion through the link system to the end of the floating lever and thus brings the pilot valve back to its normal position. Each position of the governor determines a definite position of the piston in the operating cylinder and consequently the opening of a definite number of controlling valves. In many of the large units up to 20,000 kw., these controlling valves are of the unbalanced type, as shown in Fig. 297, and vary in number from two to ten depending upon the size of the turbine and the load conditions. In base-load units ranging from 20,000 to 50,000 kw., there are but two admission valves of the balanced throttle type, Fig. 298. One of these controls the steam for normal operation and the other for overload service. The first admits steam to the high-pressure stage nozzles and the second to a lower stage.

The emergency governor, or automatic stop, consists of a ring R (Fig. 300), unevenly weighted, attached to and revolving with the shaft. At normal speeds and less, the unbalanced ring is held concentric with the shaft by helical springs S. When the speed increases to 10 per cent above normal, the centrifugal force of the unbalanced portion of the ring overcomes the spring tension, and the ring revolves eccentrically. In this position the ring strikes a trip finger and closes the main throttle valve which is of the balanced emergency type. In the newer Curtis units of large design, the fulcrum of the lever operating the main control valve, Fig. 298, is held in position by a piston supported by oil pressure. The releasing of this pressure by the emergency-governor trip immediately closes the main governor. In testing out the emergency governor, oil is discharged into chamber C,

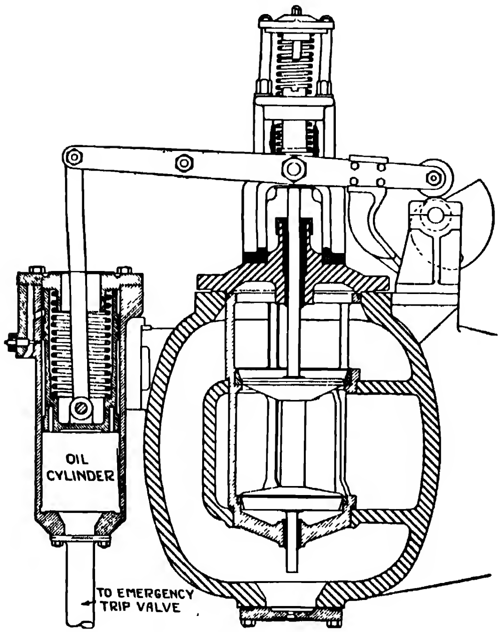


FIG. 298. Control Valve with Emergency Release.

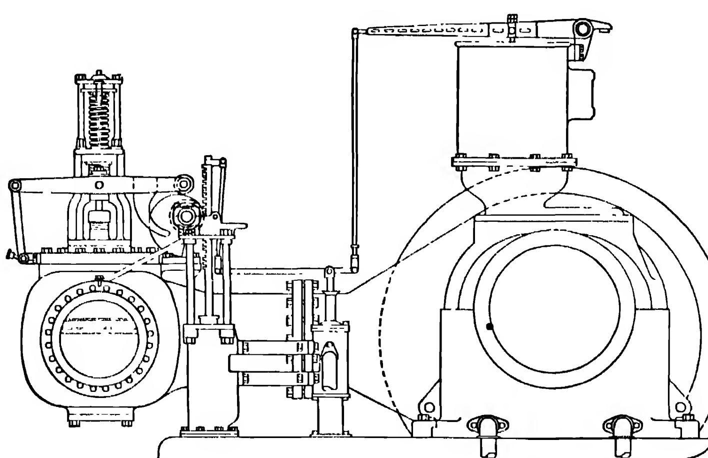


FIG 299. Governor Assembly for 35,000 kw. unit.

Fig. 300, through the agency of a hand-controlled valve, and the added weight of the oil causes the mechanism to assume an eccentric position before the actual overspeed of the turbine is reached. This trips the control valve as before. As soon as the oil supply is shut off, the oil in chamber C escapes through ports P and P and the ring again becomes balanced by the spring.

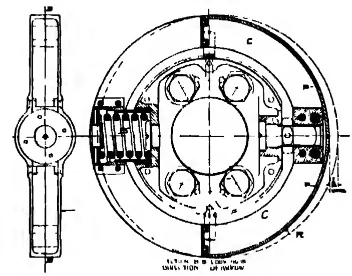


FIG. 300. Emergency Governor.

The main bearings of the turbine and generator, the thrust bearing, the flexible coupling, operating governor, and spiral gears are lubricated, and the hydraulic mechanism is operated, by oil supplied from a forced lubricating system. When the turbine runs at full speed, oil is supplied from a twin spiral gear oil pump. This pump, as well as the operating governor, is driven by a worm gear flexibly coupled to the turbine shaft.

An auxiliary turbine-driven pump furnishes oil to the turbine upon starting or when slowing down. The oil is refrigerated by circulation through a special tubular water cooler.

In the small-sized non-condensing Curtis turbines, the packing in the diaphragms separating the steam stages and in the inner sections of the high-pressure and low-pressure heads usually consists of a ring of special labyrinth packing, self-centered, with close clearance to the shaft. The high-pressure and low-pressure heads are, in addition, packed with several rings of braided, high-temperature packing. When furnished to operate with high vacuum and, in special cases, when operating non-condensing against a high back pressure, carbon packing rings are used to pack the heads. The diaphragm packing remains unchanged.

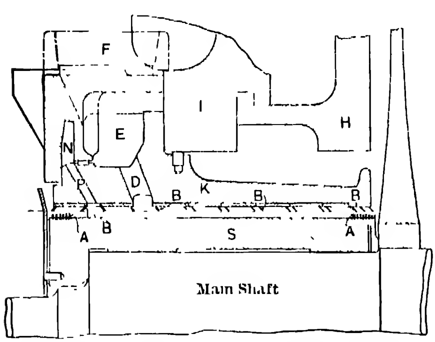


FIG. 301. Section through High-pressure Packing.

The high-pressure head of the large Curtis turbine is packed with a **labyrinth packing** of special design as shown in Fig. 301. Referring to the illustration, A . . . A and B . . . B are monel metal ribbons which are wound as a helix in a rectangular thread turned in the sleeves fitted into bushing K and in the sleeve S, respectively. The ribbons in

S are perpendicular to the axis of the machine, while those in the opposite sleeves are inclined either to the right or left, depending upon their location. This inclination is obtained by turning over the edges of the ribbon while the piece is rotating in the lathe. The port D connects with the annular opening E, and in turn exhausts through pipe connection F. The greater bulk of the steam leaking by the packing escapes through this port and through opening F. For packing in good condition, the amount of steam discharge should not exceed 3000 lb. per hr. for a 30,000-kw. machine. The steam, which is highly superheated and at atmospheric pressure, is lead to a small closed heater in which the cooling medium is the condensate from the main condenser. The rest of the packing steam passes through ports P into an annular space N and thence into a small discharge pipe leading to a small jet condenser bolted to the packing assembly on the outside of the turbine. Service water is used as the condensing medium in this condenser. Only a small quantity of water is required for this jet condenser, and the " tail " water is discharged either to waste or into a makeup storage tank. The vanes on the external packing sleeve extending from the ports D to the outside of the machine are inclined in such a direction as to resist the infiltration of air, should the jet condenser maintain a slight vacuum as is usually the case. I is the upper half of the high-pressure wheel casing, while H is the first-stage nozzle plate by means of which the entrained steam is directed upon the blades of the first-stage wheel R. The low-pressure packing is similar in design to that of the high-pressure except that steam from an external source is used for sealing. The diaphragm packing is also similar in design to that of the high-pressure packing illustrated in Fig. 301.

Figure 302 shows a horizontal section through a **Kerr "Economy" Curtis-Rateau** type turbine consisting of a single two-velocity stage Curtis element and nine Rateau stages. This machine differs from the equivalent Curtis design only in structural details and in the arrangement of the main operating governor. The governor weights, consisting of a split sleeve concentric with and mounted directly on the end of the turbine shaft, actuate a balanced-piston valve through an oil-relay system as illustrated. This type of Kerr turbine is constructed in sizes ranging from 5 to 2500 hp.

Figure 303 gives a diagrammatic arrangement of the blades and nozzles of a typical single-pressure multi-velocity-stage turbine consisting of a single set of stationary nozzles, a double row of moving vanes on a single rotor and one intermediate or reversing element.

Steam is completely expanded in the stationary nozzles P_2 and issues from the nozzles with *absolute* velocity V_1, striking the first set of moving blades at an angle α with the line of motion of the wheel. The resultant

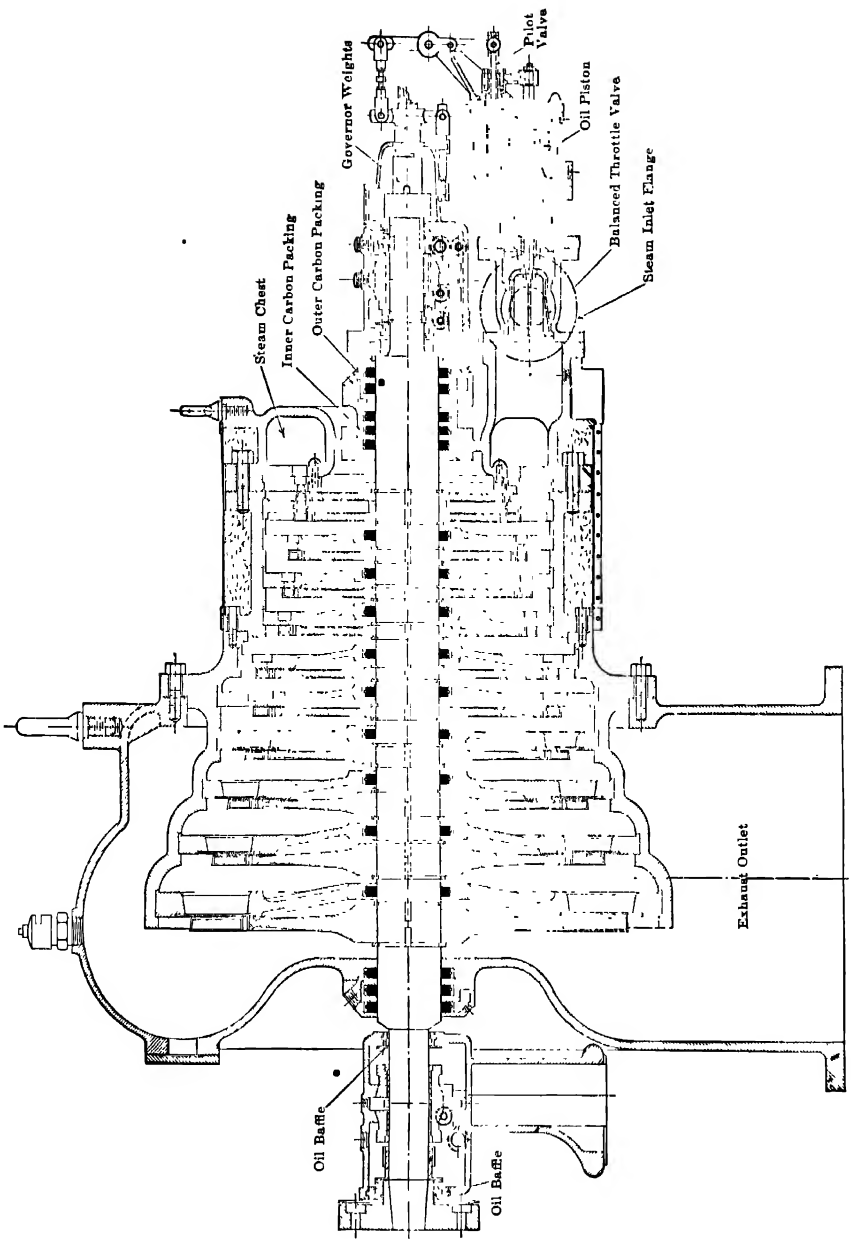


FIG. 302. Kerr "Economy" Curtis-Rateau Type Turbine.

v_1 of V_1 and the peripheral velocity u is the velocity of the steam *relative* to the vanes; and the angle β which the line v_1 makes with the line of motion of the wheel is the proper entrance angle of the blades for the first set. Neglecting friction, the exit angle γ will be the same as the entrance angle β. The resultant of v_2, the exit velocity *relative* to the blade, and u, the peripheral velocity is V_2, the *absolute* exit velocity.

Since the second set of blades is fixed and serves as a means of changing the direction of flow, the absolute velocity entering them is V_2. The angle δ formed by V_2 and the center line of the stationary blades is the proper entrance angle. Neglecting friction, the absolute exit velocity will be $V_3 = V_2$, and the exit angle will be $\epsilon = \delta$. The steam flowing from the stationary blades strikes the second set of moving blades at an angle $\epsilon = \delta$ with *absolute* velocity V_3. Combining V_3 with the peripheral velocity u, we get v_3, the velocity of the steam *relative* to the second set of moving blades. The angle θ, formed by v_3 and the line of motion of the wheel, is the proper entrance angle for the second set of moving blades. The resultant of v_4 ($= v_3$) and u is V_4, the absolute exit velocity for the first stage.

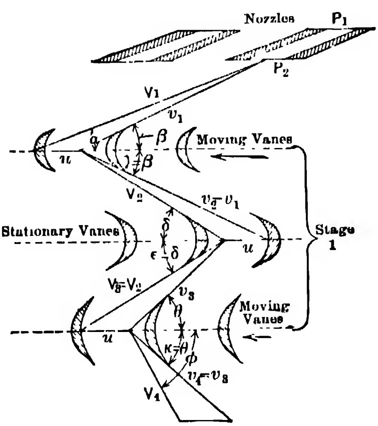


FIG. 303 Velocity Diagram, Curtis Turbine.

Any number of pressure stages may be analyzed in a similar manner; it should be noted that the initial absolute velocity of the steam entering each stage corresponds to the heat drop in the nozzle measured from the final heat content of the steam in the preceding stage to that at the mouth of the nozzle under consideration.

The "Terry Condensing," Moore "Multi-stage," and some designs of the Kerr turbine, while of the multi-pressure, multi-velocity-stage impulse type, are of composite design, consisting of a Curtis high-pressure stage and a number of Rateau intermediate and low-pressure stages. These turbines are frequently designated as Curtis-Rateau machines. By substituting a Curtis stage for several of the first Rateau stages, the windage losses are reduced and a shorter cylinder is obtained. Machines of this design are usually limited to capacities under 2000 hp.

Example 40. — A four-stage Curtis turbine develops 800 hp. on a steam consumption of 12 lb. per hp-hr.; initial pressure 150 lb. abs., superheat 100 deg. fahr., back pressure 1.5 lb. abs., peripheral velocity

450 ft. per sec., angle of the nozzle with the plane of rotation, 20 degrees. Each stage consists of two rotating elements and one stationary element. Compare the performance of the actual turbine with its theoretical possibilities.

Solution. — *Ideal Turbine.* For the sake of simplicity, it will be assumed that the final velocity of each stage is zero and that the heat drop in the first set of nozzles is one fourth of the total theoretical drop, assuming adiabatic expansion.

From steam tables $H_1 = 1249.6$ B.t.u.

From entropy tables or Mollier diagram $H_n = 934.6$.

Total heat drop $= 1249.6 - 934.6 = 315$.

Heat drop in first stage $315/4 = 78.75$.

The velocity of the jet in the first stage is

$$V_1 = 224 \sqrt{78.75} = 1985 \text{ ft. per sec.}$$

By laying off this initial velocity in direction and amount, and combining it with the peripheral velocity as in Fig. 307, the absolute velocities V_2 and V_3 may be readily obtained.

The kinetic energy absorbed in the first set of moving blades, per lb. of steam, is

$$E_1 = (V_1^2 - V_2^2) \div 2g = (\overline{1985}^2 - \overline{1170}^2) \div 64.4 = 39{,}930 \text{ ft-lb. per sec.}$$

and in the second set of moving blades

$$E_2 = (V_3^2 - V_4^2) \div 2g = (\overline{1170}^2 - \overline{670}^2) \div 64.4 = 14{,}280 \text{ ft-lb. per sec.}$$

The total energy converted into useful work is

$$39{,}930 + 14{,}280 = 54{,}210 \text{ ft-lb. per sec.}$$

Had the entire heat drop been utilized in doing work the total energy would be

$$1/64.4 \times \overline{1985}^2 = 61{,}180 \text{ ft-lb. per sec.}$$

The difference $61{,}180 - 54{,}210 = 6970$ represents the loss due to the residual velocity of the steam leaving the last bucket.

Since the steam is practically brought to rest before entering the second set of nozzles, the heat equivalent of this energy or $6970/778 = 8.96$ B.t u. increases the final heat content; thus

$$H_2 = 1249.6 - 78.75 + 8.96 = 1179.8 \text{ B.t.u.}$$

But a total heat drop per stage of 78.75 B.t.u. was assumed as a requirement of the problem, and the final result obtained above shows it to be $78.5 - 8.96 = 69.54$. By trial and adjustment or by means of empirical formulas, a value of H_2 may be obtained which will fulfill the given conditions. Such an analysis is beyond the scope of this book, and the reader is referred to Forrest E. Cardullo's article "Energy and Pressure Drops in Compound Steam Turbines," *Trans. A.S.M.E.*, vol. 33, p. 325, 1911.

The remaining stages may be analyzed in a similar manner.

It should be borne in mind that in the actual turbine the velocity will be less than the theoretical on account of frictional resistances in the nozzles and blades, and the heat content H_1, H_2 . . . H_n will be greater than that of the ideal mechanism. Radiation, leakage, windage, and other losses must also be considered in determining actual conditions.

Neglecting the residual energy in the exhaust, the total heat drop $H_1 - H_n$ is available for doing useful work, and the water rate of the ideal turbine is

$$W = 2547 \div (H_1 - H_n) = 2547 \div 315 = 8.1 \text{ lb. per hp-hr.}$$

Heat consumption per hp. per min.

$$= 8.1\ (1249.6 - 83.9)/60 = 157 \text{ B.t.u.}$$

Thermal efficiency

$$E_r = (1249.6 - 934.6) \div (1249.6 - 83.9) = 0.27.$$

Actual Turbine

Steam used per hour = 800 × 12 = 9600 lb.
Steam used per second = 9600 ÷ 3600 = 2.66 lb.
Hp. developed per lb. of steam flowing per sec. = 800 ÷ 2.66 = 300.
Kinetic energy converted into useful work:

$$300 \times 550 = 165{,}000 \text{ ft-lb. per sec.}$$

Thermal efficiency

$$E_t = 2547 \div 12\ (1249.6 - 83.9) = 0.182.$$

Heat consumption, B.t.u. per hp. per min.

$$12(1249.6 - 83.9)/60 = 233.$$

Rankine cycle ratio $= E_t/E_r = 0.182/0.270 = 0.675.$

207. Reaction Steam Turbine. — The reaction turbine is a multi-pressure single-velocity machine in which the reaction rather than the impulse of the jet is the force which drives the rotor. In this type of turbine, the expansion of the steam is subdivided into a great number of stages of small pressure drops, the steam expanding in the moving as well as in the stationary elements. Each stage consists of a row of stationary and a row of rotating vanes, the various stages being arranged in such a manner that the entire expansion resembles in effect a single divergent nozzle with the exception that the dynamic relationship of jet and vanes is productive of a comparatively low velocity from inlet to outlet. The action of the steam on the blades is illustrated in Fig. 304. Steam is expanded in the first row of stationary blades from pressure P to P_1 and accelerates the jet. The velocity of the jet issuing from these stationary nozzles is such that steam enters the adjacent set of moving blades practically without impulse. The steam expands from pressure P_1 to P_2 in passing through the first set of moving blades and exerts a reactive force on the blades.

The jet with low residual velocity is deflected from the moving blades to the entrance of the second set of stationary nozzles. In this second set of stationary nozzles, the steam is expanded from pressure P_2 to P_3 and the jet strikes the second set of moving blades at practically the same velocity as that of the blades. This process is repeated in each element of the turbine, the steam expanding as it flows from element to element in its passage to the condenser. It will be seen that the rotating force is primarily due to reaction though there may be some impulse when the jet strikes the moving members. There is no sudden change in pressure at any point; the reduction seldom exceeds 3 lb. at any row of blades, and the steam velocities therefore are comparatively low. The path of the steam from inlet to outlet may be axial (parallel to the axis) or radial (at right angles to the axis). The **Parsons** (British), **Westinghouse** (American), and **Allis-Chalmers** (American) are representative of the axial-flow type, and the **Ljunstrom** (Swedish) of the radial-flow type.

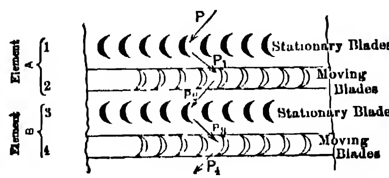


FIG. 304. Blade Arrangement, Reaction Turbine.

The Allis-Chalmers, high-pressure, single-cylinder, condensing turbine is the only American single-cylinder machine employing the straight reaction principle throughout all stages. The earlier designs of Westinghouse steam turbines were of the straight reaction type, and many of the large multi-cylinder units are of this type, but the modern high-pressure single-cylinder units of whatever size operate on the combined impulse-reaction principle.

Figure 305 shows a general assembly of an **Allis-Chalmers,** high-pressure, single-cylinder condensing steam turbine illustrating the straight reaction, axial-flow principle. The turbine consists essentially of a fixed, horizontally-split casing or cylinder and a revolving spindle or drum. The stationary blades are inserted radially into circumferential grooves in blade- or foundation-rings, which in turn are secured to the cylinder. The rotating blades are mounted in rows on separate rings or directly fitted to the spindle or drum. Each row of blades, both fixed and rotating, extends completely around the turbine, and the steam flows through the full annulus between the spindle and cylinder. Theoretically, the length of the blades and the diameter of the spindle which carries them should continuously and gradually increase from the steam inlet to the exhaust so as to accommodate the increasing volume of steam. Practically, however, the desired effect is produced by making the spindle in steps as illustrated. The blades in each step are arranged in groups of increasing

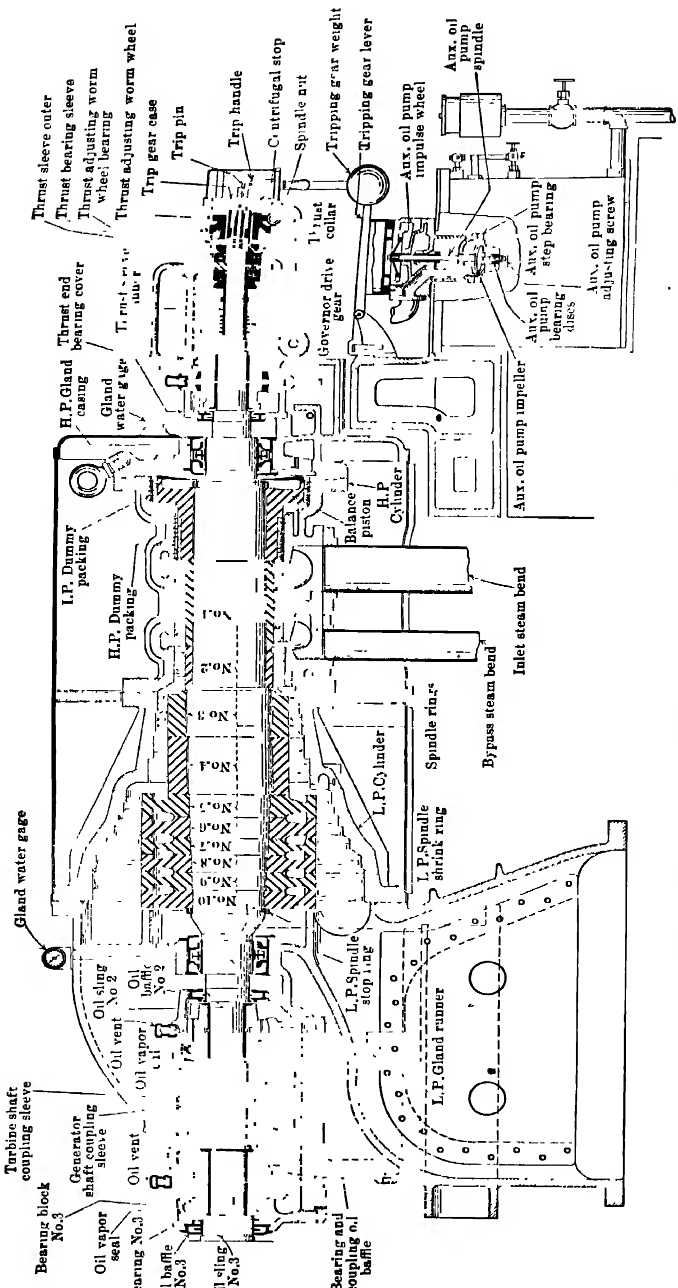


FIG. 305. Allis-Chalmers High-pressure Condensing Steam Turbine.

length. The blades are usually shorter at the beginning of each step than at the end of the preceding step, the change being made in such a way that the correct relation between blade length and spindle diameter is secured.

Because of the difference in diameter of the "steps," there is an end thrust on the spindle due to the difference in steam pressure at the end of each step. In the smaller sizes of Allis-Chalmers turbines, the thrust is neutralized by **balance pistons** mounted on the rotor and revolving inside a supplementary cylinder, or "**dummy.**" Each piston is then subjected to the same difference of pressure as the rotating drum by means of equaliz-

FIG. 306. High-pressure Balance Piston Packing (Allis-Chalmers).

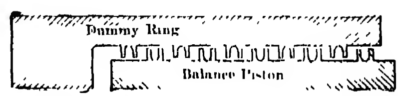


FIG. 307. Low-pressure Balance Piston Packing (Allis-Chalmers).

ing pipes. In the larger sizes, the largest piston is omitted and in its stead a smaller piston is used at the other end of the turbine, this piston having a total effective area equal to the effective annular area of the largest piston. In the latter construction, the equalizing pipe for this stage is omitted, the pressure on the balance piston being equalized with that on the third stage of the blading by means of passages through the rings on the spindle. Balance pistons and dummies do not come into contact with each other, and leakage of steam is minimized by alternate rings as shown in Figs. 306 and 307, which form a **labyrinth packing.** As a general rule all blades of 1/2 in. projected width and above are cast into foundation rings and the smaller sizes are individually formed and swaged into the foundation ring. The tips of all blades are bound together with a shroud ring, thereby insuring rigidity and accurate alignment. Blades over 3 in. in length are secured against vibration by wire lacing or stiffening strips, Fig. 308. The high-pressure end of the cylinder and spindle is sealed against leakage of steam by a **water-packed gland** which is in principle a centrifugal pump runner fixed on the turbine spindle and revolving in a casing, Fig. 309. A similar seal is used on the low-pressure end, but, owing to the high pressure at the gland due to the low-pressure balance piston, an additional seal of the labyrinth type is provided to pre-

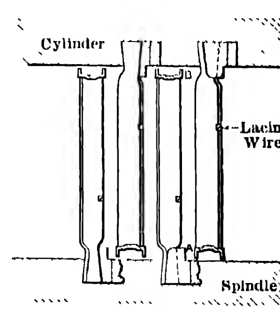


FIG. 308. Turbine Blading (Allis-Chalmers).

vent excessive leakage from the inside of the balance piston to the exhaust chamber. The main bearings are of the self-adjusting ball-and-socket type and are lubricated by means of a pump geared to the main shaft of the turbine. The oil, after it drains from the bearing, passes through a strainer into a collecting reservoir whence it is pumped through a cooler and back to the bearings. A centrifugal auxiliary oil pump is furnished, for use when starting up or stopping the turbine, or in case of emergency. The main turbine governor is of the floating-lever oil-relay type and controls the speed by throttling.

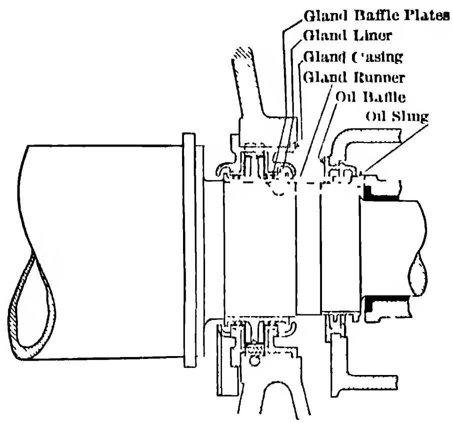


FIG. 309. High-pressure Spindle-gland (Allis-Chalmers).

Figure 310 gives a diagrammatic arrangement of the fixed and stationary blades in the first stage of a multi-stage reaction turbine. The steam enters the stationary blades at a comparatively low initial velocity and is there partially expanded and impinges against the moving blades at velocity V_1. In practice V_1 is made such that there is practically no impulse when the jet strikes the vanes. In passing through the moving vanes, the steam is further expanded and leaves at absolute

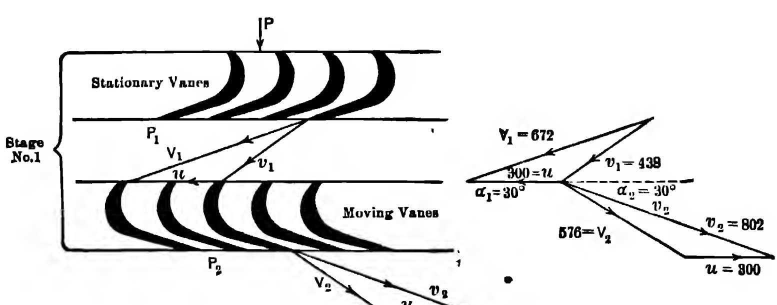


FIG 310. Velocity Diagram — Reaction Turbine.

velocity V_2, exerting a reactive force on the rotor. The steam enters the second set of stationary blades with an absolute velocity V_2 and is still further expanded to velocity V_3, and so on.

The energy, E_1, imparted to the steam in the first set of stationary

blades, considering the flow as purely adiabatic, is

$$E_1 = W(H_1 - H_2)/778 = WV_1^2 \div 2g \tag{187}$$

in which

H_1 = initial heat content, B.t.u. per lb.,
H_2 = heat content after expansion through the blades, B.t.u. per lb.,
W = weight of steam, lb. per sec.,
V_1 = velocity imparted to the jet by expansion.

The absolute spouting velocity $V_s = V_o + V_1$, in which V_o = entrance velocity to the fixed blades.

The energy, E_2, imparted to the steam in the first set of moving blades is

$$E_2 = W(v_2^2 - v_1^2) \div 2g \tag{188}$$

in which

v_1 = relative velocity of steam entering the moving blades,
v_2 = relative velocity of steam leaving the moving blades.

The total energy available in the first stage is $E_s + E_2$, in which E_s = kinetic energy of the jet leaving the stationary vanes $= WV_s^2 \div 2g$.

The total energy, E_t, converted into useful work in this stage is, therefore,

$$E_t = E_s + E_2 - WV_2^2 \div 2g$$
$$= (V_s^2 + v_2^2 - v_1^2 - V_2^2)W \div 2g \tag{189}$$

V_2 = absolute velocity of the steam leaving the moving blades. This residual velocity will also be the initial entrance velocity of the second stage.

Each stage may be analyzed in a similar manner.

Example 41. — Construct the velocity diagram and calculate the work done in the first stage of a frictionless reaction turbine for the following conditions: Heat drop per stage = 18 B.t.u. per lb. of steam; peripheral velocity = 300 ft. per sec.; exit angle = 30 deg.; entrance velocity, V_s, = 0; rate of steam flow, 1 lb. per sec.

Solution. — The velocity imparted to the steam in the first set of stationary blades is

$$V_1 = 224\sqrt{18/2} = 672 \text{ ft. per sec.}$$

The spouting velocity is

$$V_s = V_o + V_1 = 0 + 672 = 672 \text{ ft. per sec.}$$

Lay off V_s in direction and amount and combine with u = 300, Fig. 310. The resultant is v_1, the velocity of the steam relative to the blades.

The angle between v_1 and the line of motion of the wheel will be the entrance blade angle. From the diagram $v_1 = 438$. The energy given up by expansion in the moving blades is

$$E_1 = 778 \times 18 \div 2 = 7002 \text{ ft-lb. per sec.}$$

Substituting $v_1 = 438$ and $E_1 = 7002$ in equation (188), we have,

$$7002 = (V_2^2 - \overline{438}^2) \div 64.4$$

or,

$$v = 802 \text{ ft. per sec.}$$

The resultant of v_2 and u is V_2, the residual velocity of the steam leaving the moving blades. From the diagram $V_2 = 576$.

The energy converted into work in the first stage is from equation (189)

$$E_t = (\overline{672}^2 + \overline{802}^2 - \overline{438}^2 - \overline{576}^2) \div 64.4$$
$$= 10{,}420 \text{ ft-lb. per sec. for each lb. of steam flowing through the turbine.}$$

In the actual turbine the various friction and leakage losses must be included in the calculation. Such an analysis is beyond the scope of this text and the reader is referred to the accompanying bibliography.

208. Combined Impulse-reaction Steam Turbines. — The use of a single-impulse element for the first stage of the expansion in a reaction turbine is desirable in many cases, inasmuch as it replaces, without any appreciable sacrifice of economy, a considerable number of rows of blading in the least efficient stage of the reaction turbine and makes possible a shorter and consequently a stiffer rotor. The entering steam is confined in the nozzle chambers of the impulse element until its pressure and temperature have been materially reduced by expanding through the nozzles. As the nozzle chamber is cast separately from the main cylinder, the temperature and pressure differences to which the cylinder is subjected are correspondingly decreased. From 20 to 50 per cent of the total heat drop takes place in the impulse element, the exact amount depending upon the initial steam conditions.

With the exception of the impulse type described in paragraph 204, all recently constructed single-cylinder, high-pressure, Westinghouse steam turbines are of the combined impulse-reaction type. Figure 311 shows a section through a 1500-kw. turbine illustrating the usual design, and Fig. 313 shows a section through a 35,000-kw. unit representing the latest practice in large single-cylinder machines.

Referring to Fig. 311, the entire rotor, comprising the reaction spindle, impulse disc, and turbine shaft, is a single steel forging, thereby insuring great strength and rigidity. The impulse element consists of a single set of nozzles, two rows of blades on the rotor, and one set of intermediate stationary reversing blades. In this machine approximately 45 per cent of the total energy developed is absorbed in the impulse element. The

reaction element consists of 12 straight reaction stages with the rotating elements mounted on a single cylindrical drum. There are no balance pistons; dummy rings and a thrust bearing take up any unbalanced axial force that may exist. The main bearings, glands, and seals are similar to those of the larger unit described further on. The governor, Fig. 312, is driven from the main shaft through a worm gear, and acts directly on the steam-admission valve.

Referring to Fig. 313, it will be seen that the rotor of the large single-cylinder unit is composed of two forged-steel sections which are held in

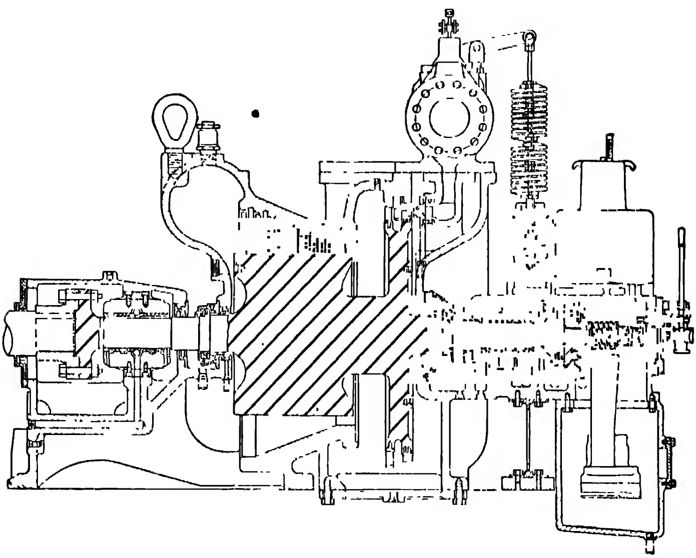


FIG. 311. Assembly of 1500-kw. Westinghouse Impulse-reaction Turbine.

place by a pressed fit and reinforced by bolts. The high-pressure section contains the rotor for the impulse element and two balance pistons, and the low-pressure section carries reaction blading only. The high-pressure casing is cylindrical in shape and houses the impulse elements and two dummy rings. The valve chest is placed above the cylinder and is rigidly attached to the casing at the primary steam inlet. The reaction casing and blade rings are shaped so as to give a diverging steam path of conical section from inlet to exhaust outlet. The exhaust chamber, furthermore, contains partitions designed on stream-line principles so as to insure uniform distribution of the exhaust steam over the whole condenser inlet opening. The first 19 rows of reaction blading are of the usual design with interlocking roots and brazed wire lashing. The rest of the blades are of the **Bauman** type, which permit of increased steam area without

increase in blade length. The Bauman principle is to divide the steam into two belts by means of partitions on the blades and auxiliary exhaust passages in the casing. The outer path is proportioned so that the steam diverted in this direction is expanded completely and efficiently. Steam of the inner path is by-passed with practically no expansion in the lower portion of the partitioned blades, but the remaining available energy is absorbed by the added row of blades. The combined length of the added

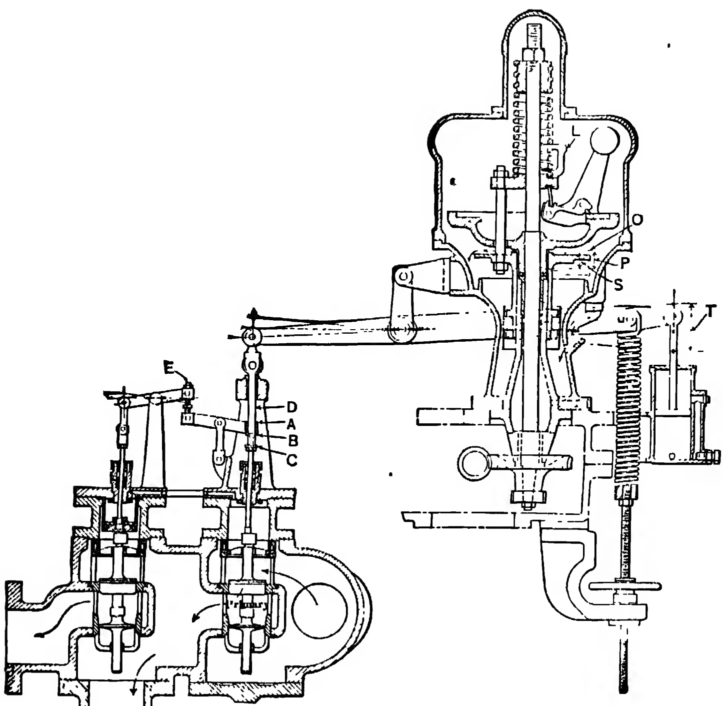


FIG. 312. Assembly of Governor Mechanism, Direct Control.

row of blades and that of the upper portion of the preceding partitioned blades is sufficient to give the desired total exhaust area without an increase in blade length. By adding one row each of revolving and stationary blades, the capacity will be increased 60 per cent above the original. In adding two rows and arranging an additional steam belt, the increase is 120 per cent, and with three additional rows, 170 per cent.

The **balance** or **dummy pistons** are connected through equilibrium pipes with suitable pressure zones in the turbine. Steam leakage past the dummies is prevented by **labyrinth packing** of the type illustrated in Fig. 314. Duplex sealing glands are provided at each end of the turbine, that

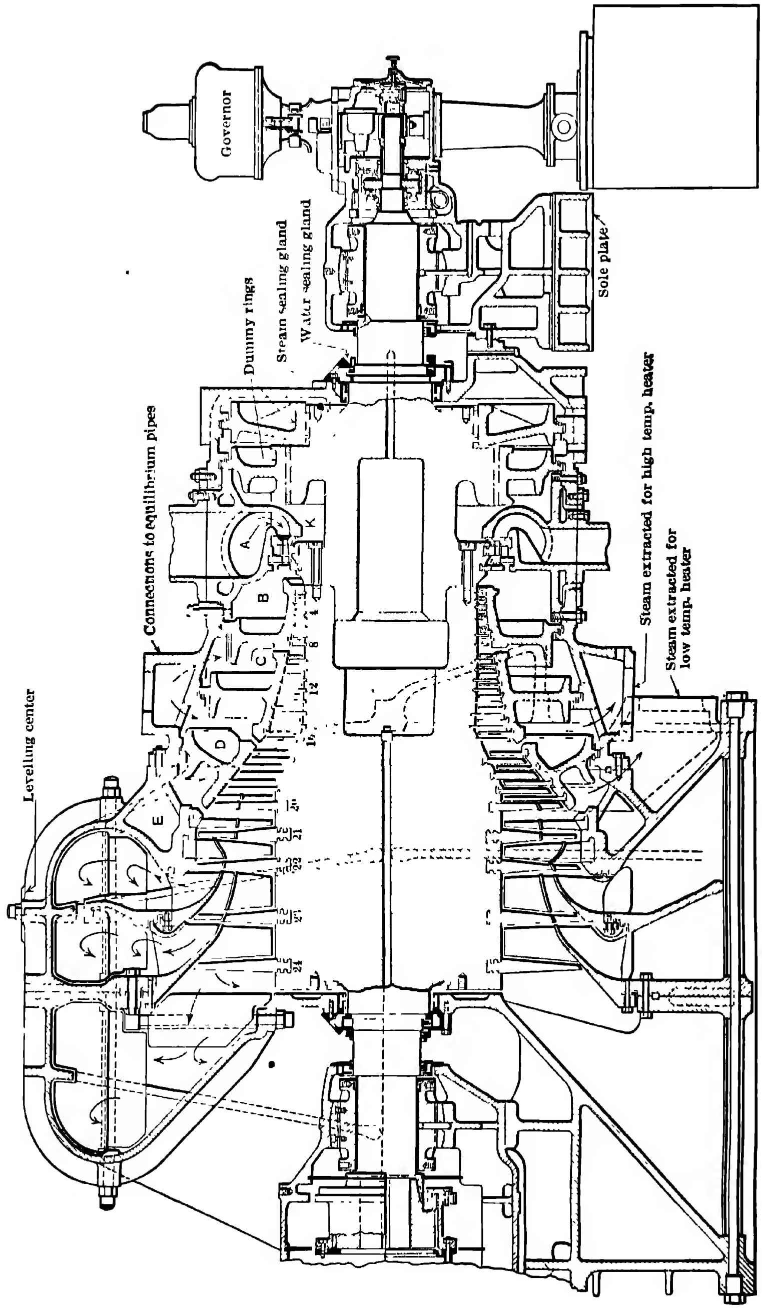


FIG. 313. Assembly of 35,000-kw. Westinghouse High-pressure Single-cylinder Condensing Turbine.

next the atmosphere being a water gland of the centrifugal impeller type, while the other is a steam labyrinth gland. The latter is useful in starting up, since the water seal does not become effective until an appreciable degree of speed is obtained. Any unbalanced thrust is taken up by a **Kingsbury thrust bearing,** Fig. 315. The main bearings are of the self-aligning type and are kept cool by a continuous flood of oil. A closed oiling system is maintained by means of a pump geared to the main shaft of the turbine. The oil, after it drains from the bearing, passes through a strainer into a collection receiver whence it is pumped through a cooler and back to the bearings.

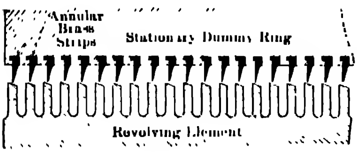


FIG. 314. Dummy Packing, Westinghouse Turbine.

Figure 316 shows an assembly of the main governor mechanism which is common to all sizes of Westinghouse reaction turbines. The movement of the governor weight is transmitted through suitable linkage to lever L, which in turn actuates rocker R. Flat-faced cam C and vibrator rod B impart a slight but continuous reciprocating motion to lever L and thus overcome the friction of rest. Rocker arm S controls a small pilot valve which admits oil under pressure to, or exhausts it from, the admission-valve operating cylinders. There are usually two admission valves, the primary and secondary, but in the large units such as illustrated in Fig. 313, there is an additional or tertiary valve. Each of these valves admits live steam to different pressure stages in the turbine. Figure 317 gives the general details of the oil relay valve gear under governor control. Rocker S, Fig. 316, controls the small pilot lever A, Fig. 317, which admits oil under pressure to, or exhausts it from, the admission operating cylinder. When oil is admitted to the operating cylinder, raising the piston, the lever C lifts the primary valve E. The lever D moves simultaneously

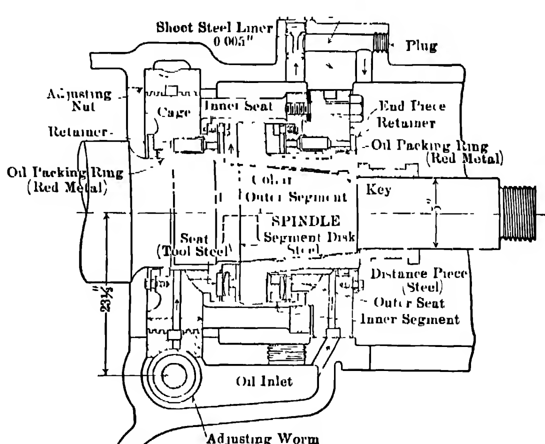


FIG. 315. Kingsbury Thrust Bearing.

with C, but on account of the valve F, the latter does not begin to lift until the primary valve is raised to the point at which its effective opening ceases to be increased by further upward travel. The secondary valve admits steam to a lower-pressure section and enables the turbine to carry a heavier load than when controlled by the primary valve alone.

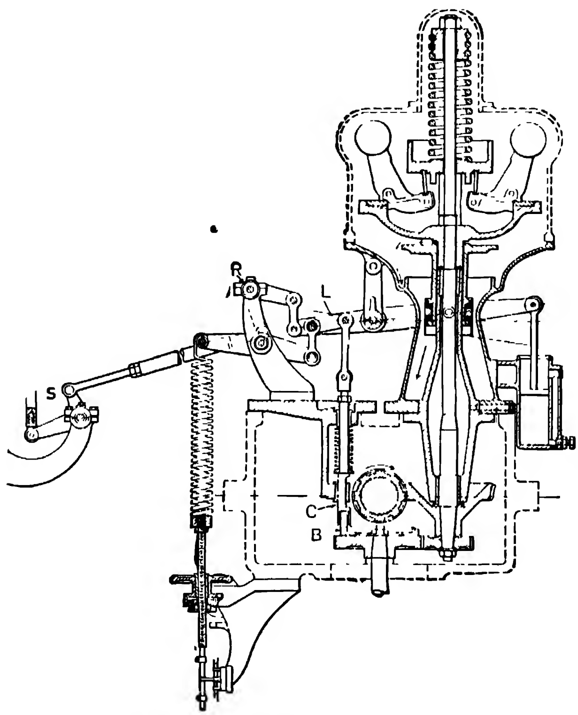


FIG. 316. Main Governor, Westinghouse Turbine.

Between the governor-controlled relay and the piston is an additional relay which is operated by a small differential piston controlled by the automatic stop governor. The descent of this piston admits full oil pressure to the under side of the piston, exhausting the upper side and so closing the steam-inlet valves regardless of the position of the governor control relay. By means of a small hand-operated plunger at the center of the shaft, the whole mechanism may be tested out, if desired, whenever the machine is shut down, without actually speeding up the turbine.

In the particular unit illustrated in Fig. 313, provision is made for extracting steam at sections B, C, D, and E. When operating at rated load with initial pressure of 300 lb. abs., 650 deg. fahr. temperature, and a vacuum of 29 in. of mercury, the pressure drops in the various sections

are approximately as follows: impulse element 300 to 120 lb. abs.; section B to C, 120 to 50 lb. abs.; section C to D, 50 to 12.9 lb. abs.; D to E, 12.9 to 3.2 lb. abs.; E to condenser 3.2 lb. abs. to 29-in. vacuum.

The double-flow type of Westinghouse turbines are no longer built and are of historical interest only.

Governing Devices of Westinghouse Geared Turbine Units: Power, Nov. 13, 1923, p. 770.

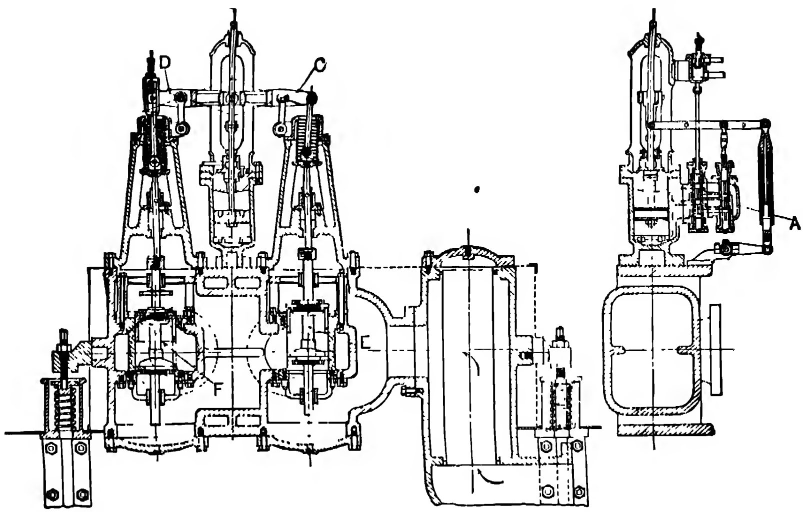


FIG. 317. Oil Relay Control, Westinghouse Turbine.

209. Compound or Multi-cylinder Turbines. — The counterflow piston engine is compounded primarily for the purpose of reducing cylinder condensation losses incident to large ratios of expansion. In the steam turbine, there is no wide fluctuation in temperature as in the counterflow engine, and hence, aside from an insignificant amount of heat lost to the surroundings, there is no cylinder condensation. For pressures under 400 lb. gage the use of more than one cylinder does not improve the heat economy of the turbine, and single-cylinder machines as high as 45,000 kw. rated capacity have the same thermal efficiency as the multi-cylinder construction for the same pressure and temperature range. Compound turbines, however, have certain structural and operating features which may prove of advantage under certain conditions. Thus, in the cross-compound design the temperature range is less in each cylinder than in a single-shell machine for the same initial and final steam conditions, and the stresses in the casing and rotor are, therefore, not so pronounced. Furthermore, the two or three cylinders are complete units in themselves, and in case of derangement or shut-down of either unit the

other may still be kept in service. The multi-cylinder design lends itself admirably to the high-pressure high-temperature reheating cycle which is now being adopted in so many new power plant projects, and a number of such units are now in course of construction. Multi-cylinder turbines of the tandem-compound and cross-compound type (two or three cylinders) are to be found in a number of plants, and a 3-cylinder 50,000-kw. unit has been installed in the Crawford Avenue station of the Commonwealth Edison Co., in which the h.p. and i.p. elements are connected in tandem to one generator and the l.p. element is a separate medium driving its own generator. The tandem-compound element is to operate at 1800 r.p.m. and the low-pressure unit at 1200 r.p.m. The 60,000-kw. units at the Seventy-fourth St. Station of the Interborough Rapid Transit Co. and at the Colfax Station of the Duquesne Light Co. are of the triple-cylinder cross-compound type consisting of one h.p. element and two l.p. elements, each driving its own generator. Multi-cylinder turbines of 25,000-kw. to 80,000-kw. rated capacity are to be found in a number of large central stations but they are not necessarily limited to large units. The De Laval Steam Turbine Co. has recently placed on the market a 6000-kw. cross-compound impulse turbine geared to direct-current generators, in which either element is capable of carrying approximately 4000 kw. in case of derangement of the other.

The Cross-compound Turbine Adaptable to a Variety of Conditions. Power, July 8, 1924, p. 50.

210. Exhaust-steam Turbines. — Exhaust-steam turbines are practically the same in design and appearance as the high-pressure turbines, except that the valves and steam passages are much larger to allow for the greater volume of the low-pressure steam. Exhaust-steam turbines use low-pressure steam only and are occasionally installed when there is an ample supply of exhaust steam to carry the load at all times. Should it be necessary to provide additional steam for an occasional failure of the main supply, high-pressure steam is furnished through a reducing valve adjusted to open only when the pressure of the exhaust falls below a predetermined amount. In installations where the supply of exhaust steam is a direct function of the load, as in connection with reciprocating engines and low-pressure turbines carrying the load in parallel, no governing mechanism is required for the turbine. Where there is no direct relation between the turbine and engine load, the usual speed-regulating governor is applied to the turbine. Straight exhaust-steam turbines are not in evidence in the modern large plant, but may be found in a number of older stations in connection with reciprocating engines, where increased capacity

was necessary and the cost of the low-pressure turbine equipment was less than that of a new unit. Low-pressure turbines have also been installed to receive the exhaust from steam hammers, rolling mill engines and other appliances using steam intermittently. In order to store up the energy during periods of excessive exhaust, and to release it during periods of diminished or interrupted supply, regenerator-accumulators have been used to advantage. The regenerator-accumulator, which is in effect a feedwater heater, absorbs the latent heat of the exhaust during the period of excess discharge and permits the heated water to vaporize at reduced pressure during the time when the exhaust supply is less than the turbine requirements. In this manner the flow of steam to the turbine is held sufficiently constant, except when the periods of regeneration are too long. Exhaust-steam turbines develop a brake hp.-hr. on a steam consumption of 25 to 35 lb., depending upon the initial steam conditions, per cent of rating carried, and the degree of vacuum maintained. By equating first cost, cost of operating, and maintenance of the low-pressure turbine equipment with the resulting heat economy effected, the net overall gain or loss over that of a high-pressure unit may be readily ascertained. Some idea of the enormous quantity of water which must be stored in a regenerator-accumulator for operating even a small turbine may be gained from the following calculations:

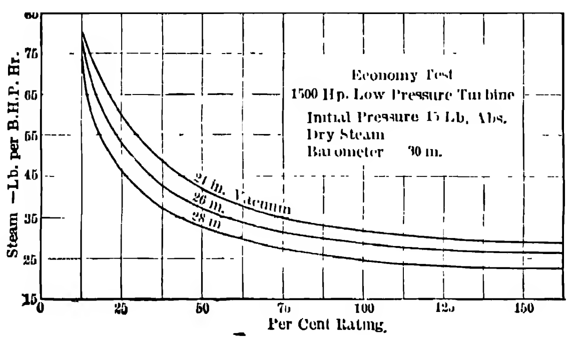


FIG. 318.

Let W = weight of water required to operate the turbine for t min., lb.

t = maximum no. of min. the exhaust supply may be entirely cut off.

s = water rate of the turbine, lb. per min.

r = mean latent heat at regenerator pressure, B.t.u. per lb.

q_1 = heat of the liquid corresponding to maximum temperature of water in regenerator, B.t.u. per lb.

q_2 = heat of the liquid corresponding to minimum temperature of water in regenerator, B.t.u. per lb.

Then $W = tsr \div (q_1 - q_2)$ (190)

If the regenerator is to absorb M lb. of exhaust steam in t min. as in case of a sudden flux of exhaust, the weight of water W_1 required is $W_1 = Mr \div (q_1 - q_2)$. (190a)

Example 42. — Determine the weight of water to be stored in a regenerator to operate a 500-hp. exhaust-steam turbine for five minutes if the steam supply is entirely cut off; pressure drop 17 to 14 lb. abs., turbine water rate 30 lb. per hp-hr.

Solution. — From steam tables and the values specified in the example, we have

$t = 5$, $s = 500 \times 30 \div 60 = 250$, $r = (965.6 + 971.9)/2 = 968.8$, $q_1 = 187.5$, $q_2 = 177.5$,

Substituting these values in equation (190) and solving

$$W = 5 \times 250 \times 968.8 \div (187.5 - 177.5) = 121,100.$$

If the regenerator is to absorb 2000 lb. of the exhaust steam in five minutes during a period of sudden flux,

$$W_1 = 2000 \times 968.8 \div (187.5 - 177.5) = 193,760.$$

Theory of Steam Accumulators and Regenerative Processes: F. G. Gasche, Proc. Eng. Soc. Wes. Penn., Dec., 1912, p. 723

Generating Power with Exhaust from Mill Engines: Power Plant Engrg., July 1, 1921, p. 641.

In small plants, where the investment cost of a regenerator would not be justified and the supply of low-pressure steam is equal to the demand of the turbine except at infrequent intervals, low-pressure turbines are installed in connection with a simple reducing valve or equivalent.

For a description of a high-pressure accumulator which has recently found application in Europe, consult *Power*, Aug. 22, 1923, p. 322.

211. Mixed-pressure Turbine. — Where the variation in supply of exhaust steam bears no relation to the various amounts required by the low-pressure turbine, the latter is usually designed to run on both high-pressure and low-pressure steam, at the same time, using all of the low-pressure steam available and sufficient supplementary high-pressure steam to carry the load. Such a combination unit is known as a mixed-pressure turbine, and has practically supplanted the straight exhaust-steam machine in all modern plants. The transition from all low-pressure to all high-pressure, through all the conditions intermediate between these extremes, is provided for automatically by the turbine governor mechanism. With this arrangement, it is not necessary for purposes of economy to proportion exactly the low-pressure turbine to the amount of exhaust steam available, but within limits it may be made as large as the load demands. Mixed-pressure turbines have been constructed in single units

as large as 10,000 kw. and have practically supplanted the straight low-pressure design in the modern industrial plant.

Mixed-pressure Turbine versus a New Steam Plant: Power, June 10, 1924, p. 934.

212. Bleeder or Extraction Turbines. — Any type of multi-pressure-stage turbine which is designed so that steam may be extracted at one or more points between the steam inlet and the exhaust outlet is usually designated as a bleeder, or extraction turbine. Evidently, the larger the number of pressure stages, the greater will be the range of steam pressures at which bleeding can be effected. The bleeder turbine may be likened to a reducing valve which furnishes lower-pressure steam from which part of the heat drop has been converted into power, instead of being dissipated at a loss. The greater the pressure drop, the greater will be the power conversion, but the heat of the bled steam available for heating purposes is not reduced proportionately. The bleeder turbine has solved, in the most satisfactory way, the problem of the heat balance in plants where, in addition to mechanical and electrical energy, low-pressure steam is required for heating buildings, heating boiler feedwater, and process work. Bleeder turbines are designed to permit any amount of extraction, from zero, or full expansion of all the steam, to 100 per cent, or partial expansion of all the steam to the point of extraction. In some designs the entire range of 0 to 100 per cent is called for; in others only a limited amount of extraction is necessary. In the average industrial plant, steam is abstracted only from the stages where the pressure is above atmospheric and then only at one point; but in many of our most recently designed central stations, extraction is made at two to four points, with one or two in the vacuum zone. See Fig. 313. The steam bled from the turbine may be controlled by hand or automatically. Hand-valve control is satisfactory if the steam bled is practically constant. If it varies considerably and at frequent or irregular intervals, automatic control is the better arrangement. Figure 319 illustrates the principles of the automatic control of a Kerr turbine.

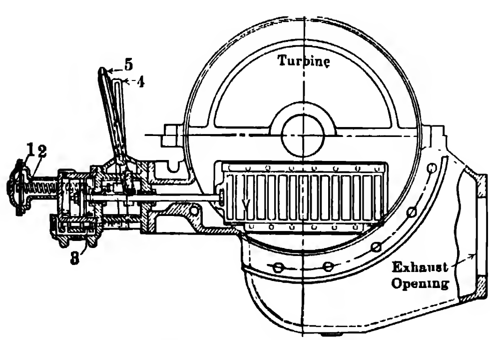


Fig. 319. Bleeder Control, Kerr Turbine.

A specially designed bleeder diaphragm equipped with a grid valve V controls the amount of steam passing through the low-pressure stages.

The movement of this grid valve is regulated by the steam pressure in the heating system, which receives the steam bled from the turbine. A small pipe connection from the receiving system to the opening marked " *l* " in the illustration acts upon a regulating diaphragm which serves as a pressure governor. Movement of this diaphragm is caused by the steam pressure and resisted by a coil spring " 2." Should the pressure in the heating system increase, the regulating diaphragm moves to the right against the pressure of the spring. The motion of the diaphragm is transmitted to a lever " 4 " pivoted at its upper end to a second lever " 5 " and having its lower end attached to an oil pilot valve " 3." Movement of the pilot valve admits oil under pressure to the pilot cylinder, which in turn moves the piston and the grid valve connected to the piston rod. This action opens the grid valve, permitting an additional amount of steam to pass through the last stages of the turbine, thereby reducing the pressure in the heating system to normal. By the connection of lever " 5 " to lever " 4," the movement of the piston rod changes the position of lever " 5," returning the pilot valve to its neutral point.

Should the pressure in the receiving system fall, owing to a sudden demand for steam, pressure on the regulating diaphragm decreases and the spring forces this diaphragm to the left. By the connection of lever " 4 " to the diaphragm, the pilot valve is moved to the right, admitting oil to the pilot cylinder and moving the piston to the left so as to close the grid valve. The closing of the grid valve permits less steam to pass through the low-pressure stages of the turbine and consequently forces more steam through the bleeder outlet into the receiving system, thereby raising the pressure to normal. The motion of the piston and lever " 5," together with increased pressure on the regulating diaphragm, returns the pilot valve to its neutral position as the pressure in the heating system becomes normal.

When the pressure in the heating system builds up to its normal point and a greater amount of steam is required to develop the rated capacity of the unit than is needed in the receiving system, the balance is passed through the last stages of the turbine, doing useful work.

The turbine is also equipped with a non-return valve installed on the bleeder outlet. Should the flow of steam from the turbine to the heating system reverse, the steam will carry the valve up against its seat and stop the flow into the turbine. If for any reason this valve should fail to close, the speed of the turbine would increase to 10 per cent over normal, at which point the emergency governor would act and forcibly close the valve.

In some cases, an automatic pressure-reducing valve is installed on the bleeder outlet to permit proper bleeder-pressure control over a very wide

capacity range. In such cases, the non-return valve described above is replaced by an oil-operated combined non-return and pressure-reducing valve, which is provided with the same non-return and emergency tripping features as previously described but with the additional reducing mechanism.

A vacuum breaker actuated by the emergency governor is installed on the exhaust end of the turbine, to prevent overspeeding due to low-pressure steam backing into the turbine.

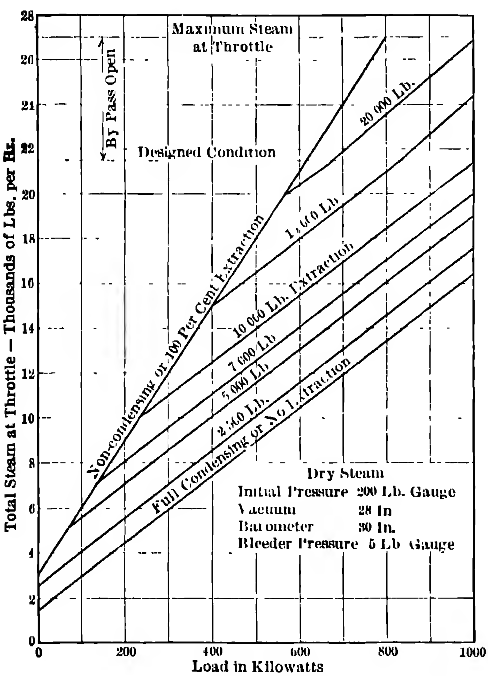


FIG. 320. Performance of 1000-kw. Bleeder Turbine.

At times when the turbine requires more steam to develop the power load than is needed in the heating system, the grid valve opens and permits low-pressure steam to pass into the vacuum stages. The extra power thus developed enables the turbine to carry the load with less steam, so that the governor shuts off a portion of the supply entering the turbine. This action of the governor, together with the admission of steam into the last stages, cuts down the quantity of steam entering the heating system and maintains it at the desired pressure.

The curves in Fig. 320 show the influence of the amount of steam extracted on the total water rate of a 1000-kw. bleeder turbine. These curves, while strictly applicable to the particular size and design tested, are typical of this class of turbine in general. The unit water rate at any load and for any amount of bleeding, and the heat content of the bled steam may be readily calculated from the diagram.

Example 43. — Using the diagram in Fig. 320, calculate the unit water rate of the turbine when delivering 500 kw. and bleeding 10,000 lb. of steam per hr. Calculate also the heat content of the bled steam.

Solution. — The total water rate for the specified conditions is found from the diagram to be 14,000 lb. per hr. Therefore, the unit water

rate is

$$14{,}000 \div 500 = 28 \text{ lb. per kw-hr.}$$

The initial heat content for saturated steam at 200 lb. gage pressure is found from steam tables to be 1198 B.t.u. per lb. The work done by 14,000 lb. of steam per hr., non-condensing, is found from the diagram to be 375 kw. The corresponding unit water rate is, therefore,

14,000 ÷ 375 = 37.3 lb. per kw-hr., and the heat content of the bled steam (see paragraph 181) is

$$1198 - 3415/37.3 = 1107 \text{ (approx.) B.t.u. per lb.}$$

When steam is extracted from a turbine at one or more stages between throttle and exhaust, it is evident that the power developed by the unit will be decreased, and in order to maintain the same output with extraction as when operating straight condensing, an additional quantity of steam must be admitted at the throttle.

If H_1 = heat content of the steam at admission, B.t.u. per lb.,

H_e = heat content of the steam at the point of extraction, B.t.u. per lb.,

H_n = heat content of the steam at exhaust.

Then $H_1 - H_n$ = heat converted to work when operating without extraction, B.t.u. per lb.

$H_e - H_n$ = heat converted to work per lb. working between the extraction stage and the exhaust.

Therefore, for every lb. of steam extracted $(H_e - H_n) \div (H_1 - H_n)$ lb. must be added to the throttle in order that the power output will remain constant. The ratio of steam added to that extracted is called the **flow factor** and may be expressed

$$F = (H_e - H_n) \div (H_1 - H_n) \tag{191}$$

While this is a simple relationship, it is difficult of application because of the variation in the values of H_1, H_e and H_n in actual practice. Any extraction or addition of steam over straight operation will alter the values of these quantities because of the changes in pressure, velocity and quality. Even when the various stage efficiencies are known for different steam conditions and loads, the determination of the heat content at the points under consideration involves laborious calculations.

Example 44. — 20,000 lb. of steam are to be extracted from the 28-lb. abs. stage of a 20,000-kw. steam turbine when operating at rated load. Determine the weight of steam which must be added at the throttle in order to develop the rated capacity under full extraction. Initial pressure 265 lb. abs., superheat 250 deg. fahr., vacuum 0.5 lb. abs. Assume that

the superheat at the point of extraction is 25 deg. fahr. and that the quality at exhaust is 0.92.

Solution. — From steam tables, $H_1 = 1340.5$, H_e for 28 lb. abs. and 30 deg. superheat = 1175.1 and H_n for 0.5 lb. abs. and 0.92 quality = 1010. Substituting these values in equation (191) and solving

$$F = \frac{1175.1 - 1010}{1340.5 - 1010} = 0.5$$

Steam to be added = 20,000 × 0.5 = 10,000 lb.

The water rate without extraction = 3415 ÷ (1340.5 − 1010) = 10.3 lb. per kw-hr.

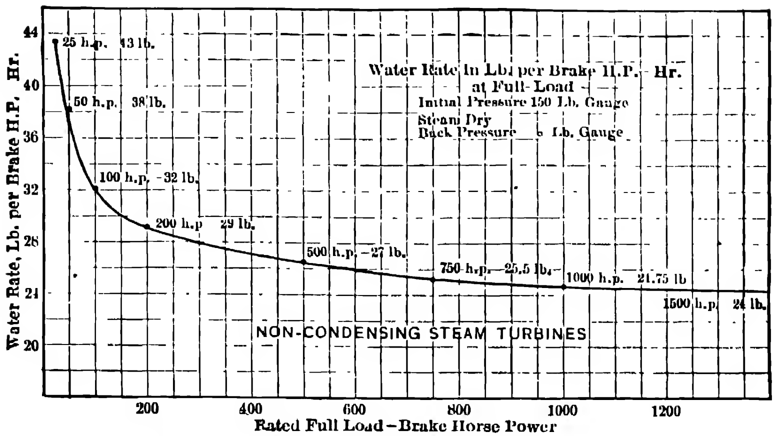
The water rate with full extraction is

(10.3 × 20,000 + 10,000) ÷ 20,000 = 10.8 lb. per kw-hr.

These values are only approximate, since the efficiencies of the various stages will vary with the amount of extraction and the addition of "make-up" steam.

Steam Bleeding and Turbine Performance Mech. Engrg., Dec. 25, p. 1144.

213. Efficiency and Economy of Steam Turbines. — A general comparison of the water rates of piston engines and steam turbines is very unsatisfactory because of the diversity in operating conditions. In a general sense the piston engine is more economical in the use of steam than the turbine for non-condensing service and the reverse is true for high-pressure, high-vacuum, condensing service. Condensing engines of the uniflow or poppet-valve type have shown superior economy (under favorable conditions) to the turbine for sizes up to 3000 hp. and in some instances up to 5000 hp., but heat economy is only one of the many factors entering into the ultimate cost of power. For high-pressure condensing service in connection with electric drives, the turbine is in a class of its own for capacities over 3000 hp., and piston engines above this size are seldom found in the modern central station. A comparison of the curves in Fig. 280, showing typical economy curves of high-speed single-valve non-condensing engines, and of Fig. 321 showing the performance of non-condensing steam turbines, is somewhat in favor of the piston engine, the difference decreasing as the size of unit increases. A similar comparison of the performance curves of compound single-valve, single-cylinder four-valve, and compound four-valve non-condensing piston engines with those of steam turbines of the same size show marked increase in economy in favor of the piston engine. For sizes between 2000 and 6000 hp., there is little difference between the steam economy of the very best grade of piston engine and that of the turbine. Piston engines above 7500 kw. have not been built for central station service; hence a comparison with the turbine for larger sizes is impossible. The Manhattan type at the



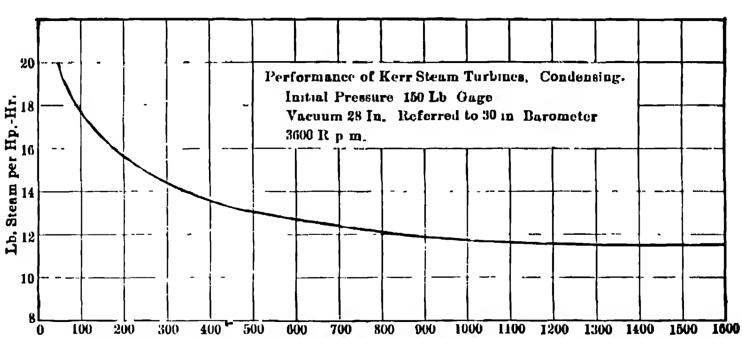
Corrections for fractional loads.—Increase full load water rate as follows: ½—20%; ¾—8%; 1—0%; 1¼—5%.

Corrections for initial pressures.—175 lb. deduct 3%; 200 lb. deduct 5%; 125 lb. add 5%; 100 lb. add 10%; 75 lb. add 20%.

Corrections for increased back pressure.—Add for each lb. back pressure 200 lb.—1%; 175 lb.—1¼%; 150 lb.—1½%; 125 lb.—2%; 100 lb.—2½%; 75 lb.—8%.

Correction for superheat.—Subtract 1% for each ten degrees superheat up to 200 degrees.

FIG. 321. Average Water Rates of High-grade Small Non-condensing Steam Turbines.



Corrections for fractional loads - Increase full water rates as follows - ½—10%: ¾—5%, 1—0%, 1¼—2.5%.

Corrections for initial pressures.- 175 lb gage deduct 2%, 200 lb. deduct 3%; 125 lb add 2½%, 100 lb. add 5%, 75 lb. add 10%.

Corrections for decreased vacuum.- 27-in add 8%, 26-in. add 15%.

Corrections for superheat.- Deduct 1% for each 14½ deg sup up to 100 deg. F; deduct 1% for each 16½ deg. from 100 to 200 deg F.

FIG. 322. Typical Performance Curves of Small Condensing Steam Turbines.

Seventy-fourth Street Station of the Interborough Rapid Transit Co. represents the largest piston engines (7500 kw.) ever constructed for central station service. The heat consumption of these engines is considerably more than that of the modern turbo-generator of the same capacity.

Except for prime movers operating on the straight Rankine cycle, water rates, or heat supplied per unit output, offer no measure of the relative heat economies since the heat added or abstracted between throttle and condenser must also be taken into consideration. In fact, the only true comparison involves the entire station heat balance and not merely the performance of the prime movers.

Thermal Efficiency of Large Steam Turbines: Power, July 29, 1921, p. 170.

Steam turbines are usually sold on a guarantee basis, that is, the particular machine in question is guaranteed to deliver the required power on a certain steam consumption under specified conditions. After being

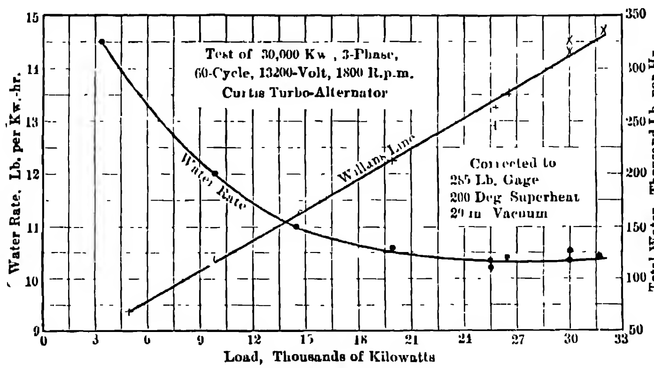


FIG. 323. Economy Test of 30,000-kw. Curtis Turbine.

installed it is frequently found that the steam conditions are different from those specified in the contract. In order to ascertain whether or not the actual performance under existing conditions meets with the guaranteed performance under contract conditions, it is customary to "correct" the test results. This correction is customarily made by finding partial corrections for pressure, superheat, and vacuum, and algebraically adding them to the test results. The partial corrections are obtained either from actual tests of machines similar in size and design to the one under consideration or indirectly from efficiency calculations. In either case the correction factors should be mutually agreed upon by the contracting parties before the acceptance tests are made. The curves in Fig. 325, though strictly applicable to the particular machine specified,

are characteristic of turbines in general and illustrate the usual form of "standard correction curves." The application of these curves is best illustrated by an example.

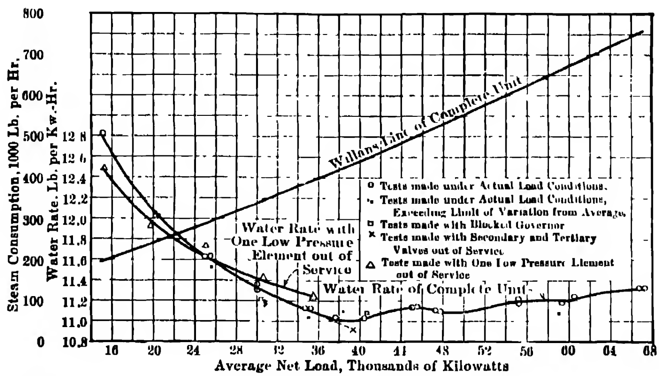


Fig. 324. Economy Test of 60,000-kw. Westinghouse Compound Turbine.

Example 45. — A 125-kw. turbo-generator is guaranteed to deliver full load on a steam consumption of 22.4 lb. per kw-hr., initial pressure 165 lb. abs., 125 deg. superheat, vacuum 28 in. referred to 30-in. barometer.

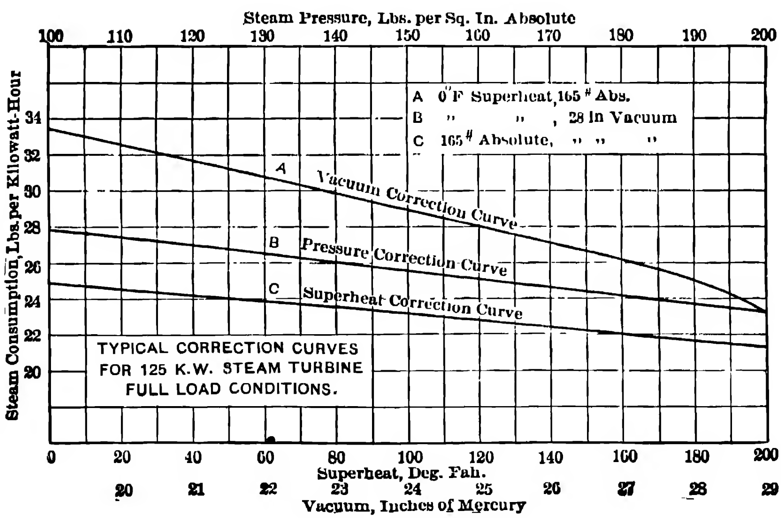


Fig. 325.

During the acceptance test the machine delivered the rated load on a steam consumption of 23 lb. per kw-hr., initial pressure 180 lb. abs., 160 superheat and 25-in. vacuum. Using the curves in Fig. 325, show whether or not the acceptance test meets with the guarantee.

Solution. — From curve *B*, we find that the steam consumption at 180 lb. pressure is 24.1 and at 165 lb. 24.7 lb. per kw-hr.; therefore, the test water rate should be increased 24.7 − 24.1 = 0.6 lb. to give the equivalent at 165 lb. From curve *C* the water rate at 160 deg. superheat is 24 lb., and at 125 deg. 22.6 lb. per kw-hr.; therefore, the test water rate should be increased 24 − 22.6 = 1.4 lb. to give the equivalent at 125 deg. From curve *B* the water rate at a 25-in. vacuum is 28 lb. and at 28-in., 25 lb. per kw-hr.; therefore, the test water rate should be decreased 28 − 25 = 3 lb. to give the equivalent at 28 in. The net corrected water rate is 23.0 + 0.6 + 1.4 − 3.0 = 22.0 lb. per kw-hr. against 22.4 lb. as guaranteed.

Manufacturers frequently use percentage correction factors similar to those printed in the legend of Fig. 325. The accuracy of the "corrected" results depends, of course, upon the care with which the factors or curves are compiled. As a rule, the corrected water rate is an approximation

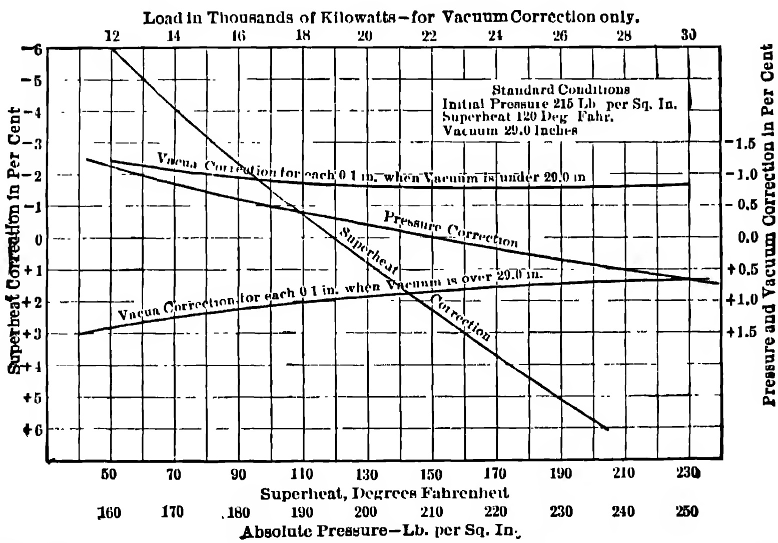


FIG. 326. Correction Factors for a 30,000-kw. Westinghouse Compound Turbine.

only, and decidedly so when test conditions depart considerably from those for which the turbine was designed.

The overload capacity of any prime mover depends entirely upon the designation of the rated load. The maximum economy of the average piston engine lies between 0.7 and full load, and for this reason the *rated* load refers usually to this maximum economical load. Evidently, if the engine is rated under its maximum possible output, it is capable of no *overload*. Under the existing system of rating, the average counterflow piston engine is capable of operating with overloads of 25 to 50 per cent, and some designs of uniflow engines as high as 150 per cent. According

to the old rating, the steam turbine was capable of overloads ranging from 100 to 200 per cent and much confusion arose in determining the station load factor. Current turbine practice gives as the normal rating the maximum continuous load which can be carried for twenty-four hours. Since all modern turbines are designed for a point of best steam consumption somewhere, regardless of what their rating may be, the *actual* rating means little.

Because of the various uses to which turbines are applied and on account of the extreme variation in design, general rules for approximating the cost of turbines are without purpose. Values based on rated capacity vary within such wide limits that average figures are apt to lead to serious error. In a general sense, steam turbines are lower in first cost than steam engines of equivalent rated capacity, irrespective of size.

Although the turbine is composed of a large number of parts as compared with a reciprocating engine of the same capacity, there are few moving parts and rubbing surfaces. The only contact between rotor and stator is in the main bearings, and the problem of lubrication is therefore a simple one. The absence of pistons, stuffing boxes, dash pots, etc., reduces the cost of maintenance and attendance to a minimum.

The floor space required by practically all types of turbines is considerably less than the space requirements of piston engines. Vertical, three-cylinder, compound, Corliss engines of the New York Edison type require the least floor space of any large slow-speed reciprocating engines, but take up about twice the space of a modern turbine installation of the same size. With non-condensing high-speed engines the comparative economy in space is less marked. The average floor space occupied by large turbine units is approximately 20 per cent that of engine units of equivalent capacity, but specific cases may be cited in which the ratio varies widely from the average. In the modern central station the actual space reduction per kilowatt of plant rating is much less than that referred to the prime mover only, because of the tendency toward less crowded conditions.

The weight of the steam turbine is very small compared with a reciprocating engine of the same horsepower. The New York Edison type of engine and generator weighs more than eight times as much as a turbine installation of equal capacity. The turbine, for this reason, and also because of the total absence of reciprocating parts, requires a relatively light foundation. In many instances the foundation consists of steel beams with concrete arches sprung between them resting upon the floor, and the basement underneath may be used for the condenser instead of the massive foundation required for the reciprocating engine.

Characteristic Curves of Steam Turbines: Power Plant Engrg., Aug 1, 1924, p. 797.

TABLE 64

GUARANTEED PERFORMANCE OF A NUMBER OF LARGE WESTINGHOUSE TURBO-ALTERNATORS

Station	No. of Cylinders	Throttle Pressure Lb. Gage	Throttle Temperature Deg. Fahr.	Back Pressure In Hg.	R p m.	Rated Capacity Kw.	Most Efficient Capacity Kw.	Water Rate Lb. per Kw.-hr.	
								Rated Capacity	Most Efficient Capacity
Barbados	1	285	617	1 5	1,800	20,000	16,000	10 84	10.75
Battle Creek.	1	200	588	1	1,800	20,000	2,000	10 65	10.65
Calumet ..	1	300	647	1	1,200	37,000	30,000	9 94	9.82
Cahokia	1	300	700	1	1,800	35,000	30,000	9.91	9.78
Colfax . .	1	265	625	1	1,800	30,000	22,500	10 25	10.24
Colfax	2	265	585	1	1,800	60,000	50,000	10 79	10.58
Crawford	3	550	725	1	1,800	50,000	50,000	7.56	7.56
Devon, Conn..	1	285	617	1.5	18,000	20,000	16,000	10.85	10.70
Grand Tower..	1	350	700	1	1,800	20,000	16,000	9 90	9.75
Hell Gate ...	2	250	607	1	12,000	40,000	28,000	10 90	10.40
Kearney..	1	325	700	1 5	1,800	35,000	30,000	9 79	9.75
Los Angeles. ..	1	350	700	1	1,800	30,000	26,250	9.49	9.44

214. Influence of Initial Pressure and Temperature. — The great majority of steam power plants are operating with steam pressures below 250 lb. per sq. in. gage and temperatures below 650 deg. fahr., and, with the exception of the large central stations, these limits are not likely to be exceeded in the immediate future. Great improvement in overall station economy has been effected under these conservative pressure and temperature conditions by eliminating many of the losses formerly considered unavoidable, and to-day we have plants which are operating continuously with overall thermal efficiencies of 19 per cent as against 15 per cent of the plant of a decade ago. The tendency in the modern large central station, however, is toward higher and higher pressures and temperatures; plants are now in operation with initial pressures of 550 lb. gage and temperatures of 750 deg. fahr. at the turbine throttle, and a number have definitely planned to use steam pressures up to 1200 lb. per sq. in. In fact, an experimental plant is under construction in England in which it is pro-

posed to generate steam at 3400 lb., somewhat above the critical pressure. A study of the Rankine cycle and the performances of the latest stations employing pressures of 400 lb. gage and temperatures of 750 deg. fahr. shows that there is little net advantage in exceeding these limits for prime movers operating on the straight Rankine cycle. With the practical temperatures limited to 750 to 800 deg. fahr., the increased range of expansion causes the steam to become saturated too early to permit the lower stages of the turbine to perform their function most advantageously. However, by reheating the steam about midway of the total expansion and by bleeding the turbine at various points for feedwater heating, the gain due to increased pressures is such as to warrant the adoption of pressures far above that found in the average plant. This intermediate reheating of the steam between stages and bleeding for heating purposes places the operation of the turbine in a cycle other than the straight Rankine.

Because of the limitation of the Rankine cycle and the impracticability of the Carnot cycle, a number of other cycles have been proposed which offer higher theoretical and probably higher commercial efficiencies than the former. At the present writing (1924) there are many revolutionary power-plant designs under consideration, and while these give promise of exceptional operating results specific data will not be available until the plants have been operated for some time. The new projects are based on the

(1) reheating cycle
(2) regenerative cycle
(3) reheating-regenerative cycle.

Reheating Cycle. — In this cycle the steam is withdrawn from the turbine after it has been partially expanded and reheated. The amount of heat added is sufficient not only to dry but to superheat the partially expanded steam, thereby increasing the hydraulic efficiency in the lower stages. The steam is then returned to the turbine and expanded to condenser pressure in the usual manner. In some of the new plants, steam is to be generated at about 550 lb. gage pressure with a temperature of about 700 to 750 deg. fahr. and passed through the high-pressure element of a compound turbine. The exhaust from the high-pressure element is reheated in a superheater, placed inside the boiler, to initial temperature, and returned to the low-pressure cylinder. In two plants now under construction, steam is to be generated at 1000 to 1200 lb. gage and 725–750 deg. temperature and expanded through a small turbine to a pressure of 300–375 lb. gage. The exhaust from this high-pressure turbine is reheated to 725–750 deg. by a superheater placed in the boiler, and then

discharged into the main turbine header. The efficiencies of the reheating cycle for various throttle pressures and reheating pressures as computed by C. F. Hirshfeld and F. O. Ellenwood are shown in Fig. 327. The curves in Fig. 327 are based on initial or throttle temperatures of 700 deg. fahr. and exhaust pressure of 1 in. Hg. abs. The points of maximum efficiency for each throttle pressure are connected by a dotted line so that the relation between throttle pressure and reheating pressure for maximum cycle efficiency are readily determined. The influence of the number of stages on the efficiency is shown by the curves in Fig. 328. These curves, while strictly applicable only to the specific conditions involved, are general insofar as the general characteristics are concerned. It will be seen that the gain for more than three stages is negligible. Calculations for the reheating cycle will be found in paragraph 403.

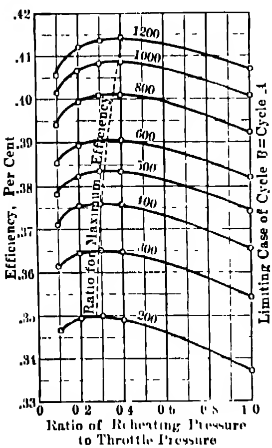


Fig. 327. Efficiency of Reheating Cycle.

Some engineers are of the opinion that the complication and expense of reheating is unwarranted and that satisfactory results may be obtained by removing part of the condensation within the dew-point stage of the turbine by means of some moisture-separating device and returning the moisture-free steam to the turbine. This solution is simple and cheap but data as to the actual performance of such an arrangement are not available.

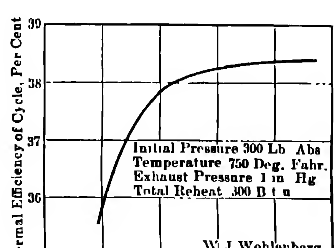


Fig. 328. Influence of Number of Stages in Reheating Cycle.

Regenerative Cycle. — In this cycle the condensate from the turbine is passed through a series of feedwater heaters in which it is heated by steam bled from different stages of the turbine from which it has just emerged as condensate. With an infinite number of such feedwater heaters, the condensate could be brought up to boiler temperature. For constructive and operative reasons, it is impractical to use more than four stages, so that the ideal cycle can only be approximated. With the present type of turbines, it appears that this approximate regenerative cycle is the best to use in large stations since it gives high heat economy, has good operating characteristics, and the first cost is not excessive. This is particularly so for pressures above 600 lb. gage. A comparison between the theoretical

thermal efficiency of the Carnot cycle, the Rankine cycle, and a regenerative cycle using saturated and superheated steam for various initial pres-

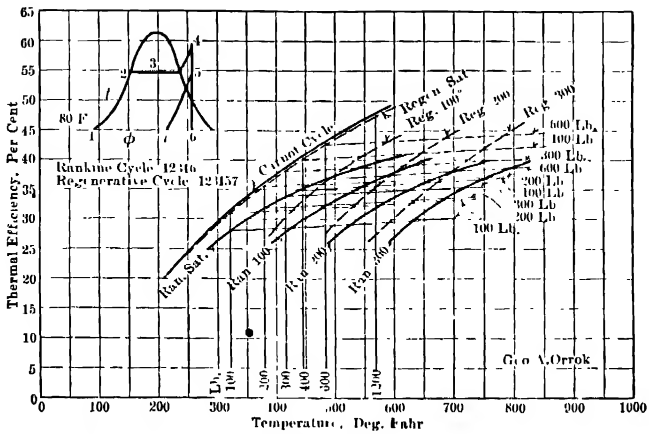


FIG. 329. Theoretical Thermal Efficiency of Various Cycles.

sures and temperatures is shown in Fig. 329. The particular regenerative cycle used in Fig. 329 is analyzed in paragraph 304. It will be seen that the regenerative cycle follows very closely the efficiency of the Carnot cycle, while the Rankine cycle falls below the Carnot cycle increasingly with the rise of pressure.

Figure 330 shows the relation between throttle pressure and temperature and the bleeding temperature to give maximum efficiency on a regenerative cycle proposed by Hirshfeld and Ellenwood (see paragraph 404). There are several stations in this country operating on a modified regenerative cycle in which the turbine is bled at one to three points, and a number are in course of construction. Calculations for the regenerative cycle will be found in paragraph 404.

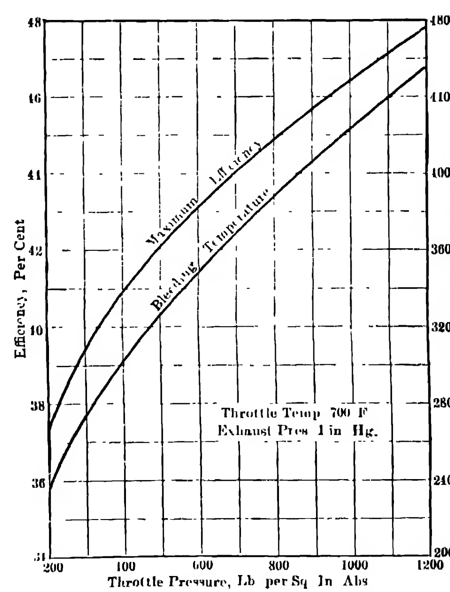


FIG. 330. Maximum Efficiency and Corresponding Bleeding Temperature of Regenerative Cycle.

Reheating-regenerative Cycle. — The bleeding process improves the efficiency by transferring energy within the cycle, and the reheating process adds energy where its availability for work is high. It would appear, therefore, that a combination reheating-regenerative cycle would have some advantages over any of the others mentioned. The curves in Fig. 331 show the relation between theoretical efficiency and initial pressure for various cycles, in which the conditions are as follows: Steam conditions A to F inclusive — initial temperature 750 deg. fahr., back pressure 1 in. Hg. Steam conditions for Curve H — initial temperature 1000 deg. fahr., back pressure 1 in. Hg.

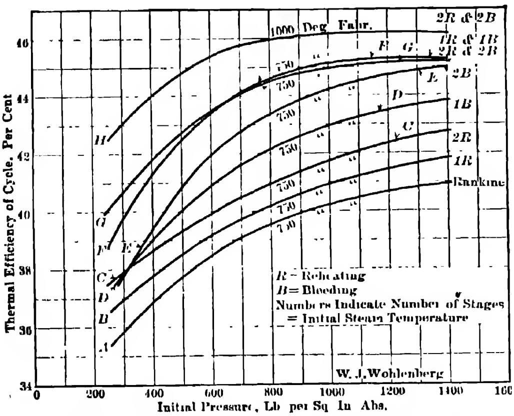


FIG. 331. Thermal Efficiency of Various Cycles

Bleeding points — 240 deg. fahr. or 25 lb. in one-stage cycles; 280 and 180 deg. fahr. or 50 lb. and 7.5 in. Hg. in two-stage cycles. The probable turbo-generator efficiencies or efficiency ratios for the various cycles with varying pressures are shown in Fig. 333. It will be noted that the efficiency decreases with the increase in initial pressure for all cycles but is less pronounced with the reheating cycle.

Reheating in Central Stations. W. J. Wohlenberg, Mech. Engrg., May 1924, p. 259.

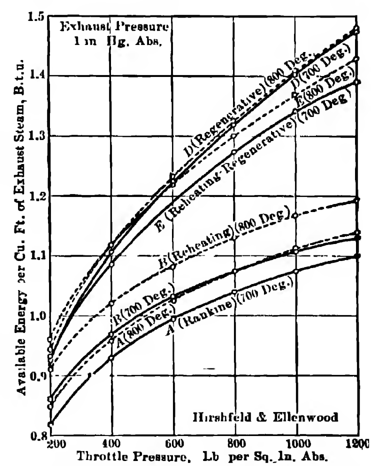


FIG. 332. Available Energy per Cu. Ft. of Exhaust Steam.

The available energy per cu. ft. of exhaust steam is important to engineers, in that it is a partial measure of the relative sizes and costs of turbines required to give the same power when operating on the different cycles. The curves in Fig. 332 show the superiority of the regenerative cycle for the conditions indicated.

For the ordinary low-load-factor plant, it is doubtful if pressures and temperatures higher than those now in use will be commercially more economical, because of the increase in investment and plant complication; but for base-load plants greater investment is justified and many of the present revolutionary designs may be standard practice in the not distant future. See also paragraph 372.

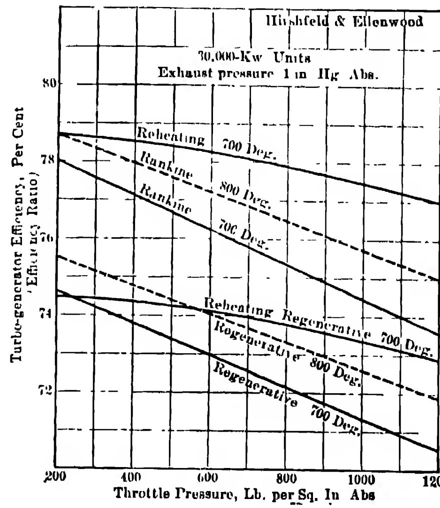


FIG. 333. Probable Turbo-generator Efficiency (Efficiency Ratio) for Various Cycles.

215. Influence of High Vacua. — The possible economy of the reciprocating engine is greatly restricted by its limited range of expansion. Cylinders cannot be profitably designed to accommodate the rapid increase in the volume of steam when expanded to very low pressures. For example, the specific volume of 1 lb. of steam under a vacuum of 29 in. (referred to a 30-in. barometer) is about 667 cu. ft., or nearly double its volume under a vacuum of 28 in. Usually the exhaust is opened at a pressure of 6 or 8 lb. abs. and consequently a large proportion of the available energy is lost. The lower vacuum in the exhaust pipe, therefore, serves only to diminish the back pressure and does not affect the completeness of expansion. Even if it were practical to expand to 1 lb. abs., the increased condensation in the reciprocating engine would probably offset any gain due to expansion unless the steam were highly superheated. A study of a number of tests of reciprocating engines shows but a slight improvement in overall plant economy due to increasing the vacuum beyond 26 in. Tests of steam turbines show a decrease in steam consumption of about 5 per cent for each inch of vacuum between 25- and 27-in. vacuum, 6 per cent between 27- and 28-in. and 8 to 12 per cent between 28- and 29-in. These values are approximate only, since the influence of vacuum on the steam consumption varies greatly with the type and size of turbine.

Since the volume of the steam increases very rapidly with the decrease in back pressure, the corresponding capacity and power required by the air and circulating pumps becomes proportionately larger. There is consequently a point where the improvement in steam economy fails to

exceed the increased power demanded by the auxiliaries. This is illustrated graphically in Fig. 334. The values in Fig. 334 refer to a specific case only, but the general principle is the same for all conditions. In the older types of condensing equipment, the cost of maintaining the vacuum above 27 in., referred to a 30-in. barometer, increased very rapidly with

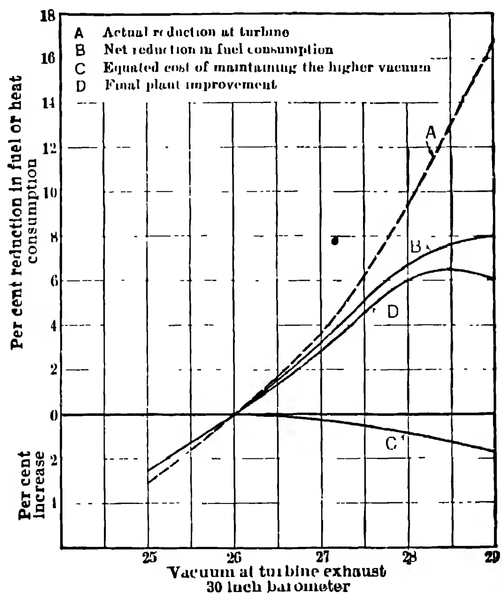


FIG. 334. Influence of Vacuum on Cost of Power.

the increase in vacuum. In the modern plant, vacua amounting to 97 per cent of the theoretical maximum (as determined by the temperature of the cooling water) are readily maintained without excessive cost. This influence of vacuum on the economy of a 30,000-kw. turbo-alternator is shown in Fig. 334.

Modern Tendencies in Steam Turbine Plants: Mech Engrg., Oct. 1924, p. 577.

50,000 Kw. Compound Parsons' Turbine for the Crawford Avenue Station: Power, Nov. 4, 1924, p. 728.

Considering the thermal efficiency of the steam-turbine electric unit as the ratio of the heat equivalent of the energy delivered by the generator to the busbars, to the heat content of the steam supplied to the turbine less the heat content of the condensate returned to the boilers, the progress made in the efficiency of such units in the United States from 1903 to 1924 is substantially as follows:

Index	Year	Size and Type of Turbo-generator Units	Initial Pressure, Lb. Gage	Temp at Throttle, Deg. Fahr	B.t.u. per Kw-Hr.	Thermal Eff. of Unit, Per Cent
A	1903	5000-kw. No bleeding; no reheating.	175	378	23,500	14.5
B	1914	20,000-kw. No bleeding, no reheating.	200	588	14,500	23 6
C	1923	30,000-35,000-kw. No bleeding; no reheating. Average good practice	230	625	13,350	25.6
D	1923	30,000-45,000-kw. No bleeding; no reheating. Most efficient stations	250	650	12,500	27.2
E	1924	30,000-50,000-kw. Bleeding but no reheating	375	700	11,200	30 5
F	1924	35,000-60,000-kw. Bleeding; single stage reheating	550	725	10,300	33 1
G	?	Proposed new stations, 100,000-kw. Single reheating	550	725	10,100*	33 8*
H	?	Proposed new stations, 100,000-kw. Two reheatings	1200	725	9,000*	38 *

Considering the thermal efficiency of the entire plant as the ratio of the heat equivalent of a kw-hr. (3415 B.t.u.) to the heat value of the coal consumed per kw-hr., the improvement in overall plant efficiency during the past 20 years is substantially as follows:

Index	Year	Plant Efficiency, Per Cent	Lb. of 14,000 B.t.u. Coal Per Kw-Hr.	Index	Year	Plant Efficiency, Per Cent	Lb. of 14,000 B.t.u. Coal per Kw-Hr.
A	1903	9 2	2 64	E	1924	21 7	1 13
B	1914	15 5	1 57	F	1924	23 6	1 04
C	1923	17 1	1 43	G	?	24 1*	1 01*
D	1923	19	1 28	H	?	26 6*	0.91*

* Expected Results

PROBLEMS

1. Steam expands adiabatically in a frictionless nozzle from an initial pressure of 200 lb. per sq. in. abs., superheat 200 deg. fahr., to a back pressure of 1 in. abs., weight discharged 7200 lb. per hr.; required:

a. Velocity of the jet at the throat.
b. Maximum spouting velocity.
c. Diameter of the throat.
d. Diameter of the mouth.
e. Quality of the steam at the mouth.

2. If the jet in Problem 1 impinges tangentially against a set of moving vanes and leaves them with residual velocity of 500 ft. per sec. required:

a. Velocity of the vanes, neglecting all friction and leakage losses.
b. Horsepower imparted to the rotor.
c. Force exerted against the vanes.
d. Water rate, lb. per hp-hr.

3. Same conditions and requirements as in Problems 1 and 2 except that the energy-efficiency of the nozzle is 94 per cent and the loss of energy between inlet and exit of the vanes is 15 per cent of the total heat drop.

4. If the nozzle in Problem 1 is to be used in connection with a multi-pressure steam turbine, required the theoretical number of stages necessary for a peripheral velocity of 500 ft. per sec. Jet impinges tangentially against the rotor and all of the available energy is absorbed in driving the rotor.

5. A single-stage impulse turbine (De Laval type) develops 200 hp. under the following conditions: Initial pressure 153 lb. abs., back pressure 4 in. abs., superheat 50 deg. fahr., water rate 14.4 lb. per hp-hr., nozzle angle 20 deg., peripheral velocity of the rotor 1200 ft. per sec. Required:

a. Thermal efficiency.
b. Rankine cycle ratio.
c. B.t.u. per hp. per min.

6. Construct the theoretical velocity diagram for the conditions in Problem 5 and sketch in the blade outlines.

7. Construct the theoretical velocity diagram for a 750-hp., 2-stage Curtis turbine operating under the following conditions: Initial pressure 175 lb. abs., superheat 150 deg. fahr., back pressure 2 in. abs., Rankine cycle efficiency 65 per cent, nozzle angle 20 deg., peripheral velocity 500 ft. per sec. Each stage consists of two rotating elements and one stationary element.

8. Construct the velocity diagram and calculate the work done per stage in a frictionless reaction turbine for the following conditions: Heat drop per stage, 16 B.t.u. per lb. of steam, peripheral velocity to be the maximum theoretically possible for the given conditions, exit angle 30 deg., entrance angle 0.

9. Determine the weight of water to be stored in a regenerator to operate a 1000-hp. exhaust steam turbine for 6 minutes if the steam supply is entirely cut off; pressure drop 15 to 12 lb. abs., turbine water rate 28 lb. per hp-hr.

10. A 300-kw. non-condensing turbine operating on steam superheated 100 deg. fahr., 150 lb. gage initial pressure and 2 lb. gage back pressure furnishes current for power and lighting. If the Rankine cycle ratio at full load is 45 per cent (e.hp.basis) and the no-load steam consumption is 12 per cent of that at full load, determine the probable water rate at one-half, three-quarters and full load operation of the unit.

11. Ten thousand lb. of steam are to be extracted from the 18-lb. abs. stage of a 20,000-kw. turbine when operating at full load. If the initial pressure is 200-lb. gage, superheat 150 deg. fahr. and vacuum 28 in., determine the weight of steam which must be added to the throttle in order to develop rated capacity under full extraction. Assume the quality at extraction to be 10 per cent higher than for adiabatic expansion and that at exhaust, 15 per cent.

12. Approximate the water rate of the turbine with full extraction.

For problems on the reheating and regenerative cycles, see end of Chapter XXIII.

CHAPTER XII

CONDENSERS

216. General. — The primary object of condensing is the reduction of back pressure, although the recovery of the condensate may be of equal importance. If a given volume of saturated steam be confined in a closed vessel, abstraction of heat will result in condensation of part of the vapor with a corresponding drop in temperature and pressure. The greater the amount of heat abstracted, the greater will be the amount condensed, and the lower will be the temperature and pressure. All of the vapor can never be condensed in practice, since this would necessitate a lowering of the temperature to absolute zero, or 492 degrees below the fahrenheit freezing point; consequently, the pressure can never be reduced to zero. With water as the cooling medium, the minimum temperature to which the vapor can be reduced is 32 deg. fahr., corresponding to a pressure of 0.0886 lb. per sq. in. or 0.1804 in. of mercury. This represents, therefore, the lowest condenser pressure possible in practice. Condensing results in reduction of pressure only when the vapor is contained in a closed vessel. Thus if the vessel is open to the atmosphere heat abstraction will result in condensation, but the pressure will not fall below that of the atmosphere.

The standard atmospheric pressure at sea level and at latitude 45 degrees is 14.6963 lb. per sq. in., corresponding to a mercury column 29.921 in. in height, temperature of the mercury 32 deg. fahr. For any other temperature there will be a corresponding height of column because of the expansion or contraction of the mercury. Steam tables are based on a standard pressure of 29.921 in. of mercury at 32 deg. fahr. and for this reason it is convenient to transfer the observed barometer and mercurial vacuum gage readings to the 32-degree standard.

The mercury column correction for any change in temperature may be closely approximated by the equation

$$h = h_1 \left[1 - 0.000101\,(t_1 - t)\right] \qquad (192)$$

in which

h = height of mercury column corrected to temperature t,
h_1 = observed height of mercury column,

t_1 = observed temperature of mercury column,
t = temperature to which column is to be referred.

Example 47. — If the height of mercury in a vacuum gage is 28.52 in., temperature 80 deg. fahr., and the barometer column is 29.85 in. in height, temperature 62 deg. fahr., transfer the readings to the 32-degree standard.

Solution. — For the barometer:

$$h = 29.85\ [1 - 0.000101\ (62 - 32)] = 29.77 \text{ in.}$$

For the vacuum gage:

$$h = 28.52\ [1 - 0.000101\ (80 - 32)] = 28.37 \text{ in.}$$

Absolute back pressure = 29.77 − 28.37 = 1.40 in.
Vacuum referred to 32-deg. standard = 29.92 − 1.40 = 28.52 in.

In condenser work, it is common practice to refer the reading of the vacuum gage to a 30-in. barometer, in which case it is necessary to increase the standard temperature of the mercury to such a figure as will increase the height of the barometer from 29.921 to 30-in.; viz., 58.15 deg. fahr. Thus, if the barometer and vacuum gage readings are corrected to a temperature of 58.15 deg. fahr. the difference between the figures will give the absolute pressure in in. of mercury at 58.15 deg. fahr., and if the difference is subtracted from 30 in. the result will give the inches of vacuum referred to a 30-in. barometer. According to A.S.M.E., 1915 Power Code, a 30-in. barometer refers in round numbers to a standard atmosphere with mercury at an ordinary temperature of 68 deg. fahr.

TABLE 65

PRESSURE OF AQUEOUS VAPOR

IN. OF MERCURY REFERRED TO 30-IN. BAROMETER

Deg. Fahr.	0	1	2	3	4	5	6	7	8	9
30			0 181	0.188	0 196	0 204	0 212	0 221	0 229	0.239
40	0 248	0 258	0 268	0 279	0 290	0 301	0 313	0 325	0 337	0.350
50	0 363	0 377	0 391	0 406	0.421	0 437	0 453	0 470	0 487	0 505
60	0 523	0 542	0 561	0 581	0.602	0 624	0 646	0 669	0 692	0.716
70	0.741	0 766	0 792	0 819	0.847	0 875	0 905	0.935	0 966	0.998
80	1.032	1 066	1 101	1 137	1.174	1 212	1.251	1 292	1 334	1.376
90	1 420	1 465	1 511	1 559	1.608	1 659	1 710	1 763	1 817	1 873
100	1 93	1 99	2 05	2 11	2.17	2 24	2 31	2 38	2 45	2.55
110	2.59	2.67	2 75	2 83	2 91	2 98	3 08	3 17	3.26	3 35
120	3 45	3 54	3 64	3 75	3 85	3·96	4 07	4 18	4 29	4 41
130	4.52	4 65	4 77	4 90	5 03	5 17	5 30	5 44	5.59	5 74
140	5.89	6 04	6 19	6 36	6 53	6.69	7 04	7.22	7.40	7.58

Example 48. — Height of mercury in vacuum gage 28.52 in., temperature of mercury 80 deg. fahr., barometer 29.85 in., temperature 42 deg. fahr.; determine the vacuum referred to a 30-in. barometer.

Solution. — For the vacuum gage

$$h = 28.52\ [1 - 0.000101\ (80 - 58.15)] = 28.46 \text{ in.}$$

For the barometer

$$h = 29.85\ [1 - 0.000101\ (42 - 58.15)] = 29.9 \text{ in.}$$

Absolute pressure in in. of mercury at temperature 58.15 deg. fahr. = 29.9 − 28.46 = 1.44 in.

Vacuum referred to 30-in. barometer = 30 − 1.44 = 28.56 in.

According to **Dalton's Laws**: (1) The mass of a given kind of vapor required to saturate a given space at a given temperature is the same whether the vapor is by itself or associated with vaporless gases; (2) the maximum tension of a given kind of vapor at a given temperature is the same whether it is by itself or associated with vaporless gases; (3) in a mixture of gas and vapor the total pressure is equal to the sum of the partial pressures. The final pressure P_c is therefore the combined pressure of the air P_a and that of the vapor P_v, or, assuming complete saturation,

$$P_c = P_a + P_v \qquad (193)$$

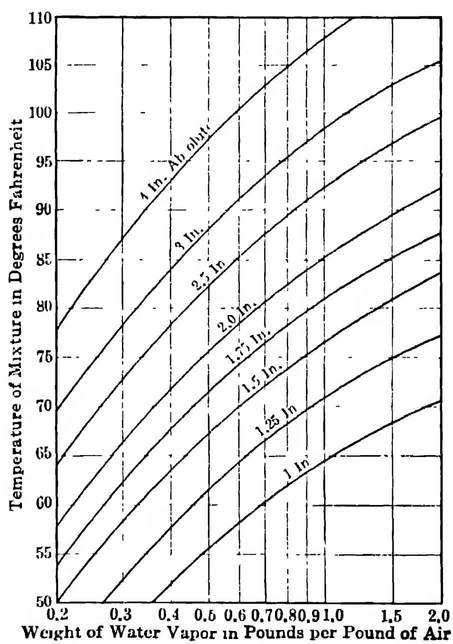


FIG. 335. Weight of Water Vapor in 1 lb. of Dry Air at Various Vacua and Temperatures.

Assuming that volume, pressure, and temperature of air and water vapor under atmospheric and condenser conditions follow the law of the ideal gas, we have

$$P_aV_a/T_a = \text{constant} = 0.754 \qquad (194)$$

in which

P_a = pressure of the dry air, in. of mercury at 32 deg. fahr.,
V_a = volume of 1 lb. of dry air, cu. ft.,
T_a = absolute temperature of the dry air, deg. fahr.

Combining equations (193) and (194) and transposing, we have

$$V_a = 0.754\ T_a/(P_c - P_v) \tag{195}$$

According to Dalton's law, V_a is also the volume of 1 lb. of dry air when saturated with water vapor under condenser pressure P_c. In other words, V_a is the volume of air-vapor mixture at pressure P_c which must be exhausted in order to remove 1 lb. of dry air. Knowing the density and pressure of the vapor content, it is a simple problem to calculate the weight of vapor contained in 1 lb. of dry air. The curves in Fig. 335 were calculated in this manner and serve to visualize this relationship.

Example 49. — If the absolute pressure in a condenser is 4 in. and the temperature of the air-vapor mixture is 100 deg. fahr., calculate the percentage of air by weight in the mixture.

Solution. — From steam tables, P_v at 100 deg. fahr. = 1.93 in., and the corresponding density is 0.00285 lb. per cu. ft.

Substituting $P_v = 1.93$, $P_c = 4.00$ and $T_a = 560$ in equation (195) and solving

$$V = (0.754 \times 560) \div (4.00 - 1.93) = 204$$

The corresponding density is 1 ÷ 204 = 0.00491 lb. per cu. ft.

Let v = volume of the condenser chamber, cu. ft.

Then the total weight of the mixture is

$$0.00491v + 0.00285v = 0.00776v$$

And the percentage of air in the mixture is

$$0.00491v/0.00776v \times 100 = 63.2 \text{ per cent.}$$

Example 50. — If the temperature within a condenser is 110 deg. fahr., and there is entrained with the steam 0.2 lb. of air per lb. of steam, required the maximum degree of vacuum obtainable.

Solution. — One lb. of saturated steam at a temperature of 110 deg. fahr. occupies a volume of 265.5 cu. ft. The corresponding vapor tension is 2.589 in. of Hg. This must also be the volume occupied by 0.2 lb. of air mixed with it, and the temperature of the air is that of the vapor (110 deg. fahr.). Then from equation (194),

$$P_a = \frac{0.754(110 + 460)}{265.5 \div 0.2} = 0.324 \text{ in. of Hg.}$$

From equation (193),

$$P_c = 0.324 + 2.589 = 2.913 \text{ in. of Hg.}$$

And the vacuum

$$= 29.921 - 2.913 = 27.01 \text{ in.}$$

If the temperature within the condenser in the preceding example were 80 deg. fahr., the pressure of the air would be 0.129 in. of mercury, and that of the vapor would be 1.031 in. Evidently the cooler the air-vapor mixture, the better will be the degree of vacuum. While it is desirable to cool the air and water vapor as much as possible, the condensate should be returned to the boiler at the highest possible temperature. In modern condenser practice this is accomplished by withdrawing the air and condensate separately, the former after it has been cooled by contact with the coolest tubes, the latter with as little contact as possible after condensation has occurred.

A condenser is a device in which the process of condensation and subsequent removal of the air and condensed steam is continuous, the degree of vacuum obtained depending upon the tightness of valves and joints, the quantity of entrained air, and the temperature to which the condensed steam is reduced.

The degree of vacuum may be expressed in different ways. (1) Excess of the atmospheric pressure over the observed vacuum. For example, a 26-in. vacuum implies that the pressure of the atmosphere is 26 in. of mercury above the pressure in the condenser. (2) Per cent of vacuum, by which is meant the ratio of the observed vacuum to the atmospheric pressure. Thus, with the barometer standing at 30 in., a vacuum of 26 in. may be expressed as $100 \times 26 \div 30 = 86.6$ per cent vacuum. This method of expression gives an idea of the efficiency of the condensing system. For example, the degree of vacuum indicated by 26 in. would be 93 per cent with a barometric pressure of 28 in., but only 84 per cent when the barometer reads 31 in. (3) Absolute pressure. Thus, a 26-in. vacuum referred to a 30-in. barometer would be indicated as a pressure of $30 - 26 = 4$ in. abs., or 1.99 lb. per sq. in. Preference is given to the last method.

The place of measurement of the vacuum should be stated, since the lowest back pressure will be found at the air-pump suction, a higher pressure in the body of the condenser, and the highest at the prime mover exhaust nozzle.

The Meaning of Atmospheric Pressure: Power, Nov 20, 1923, p. 811.

217. Effect of Aqueous Vapor upon the Degree of Vacuum. — The futility of attempting to better the vacuum by exhausting the vapor is best illustrated by a specific example.

Example 51. — Required the volume of aqueous vapor to be withdrawn per hr. from a condenser operating under the following conditions, in order that the vacuum may be increased 1 lb. per sq. in.: Temperature of discharge water 125 deg. fahr.; correspondingly vapor tension 4 in. of mer-

cury; barometer 30 in.; relative vacuum 26 in.; engine 100 hp.; steam consumption 20 lb. per hp-hr.; cooling water 25 lb. per lb. of steam condensed.

Solution. — $100 \times 20 \times 25 = 50{,}000$ lb. of cooling water per hr.
$= 833$ lb. of cooling water per min.

Now, to increase the vacuum 1 lb. per sq. in., approximately 2 in. of mercury, the temperature of the water must be lowered to 102 deg. fahr., that is, $833\ (125 - 102) = 19{,}159$ B.t.u. must be abstracted from the water in 1 min., or $19{,}159/1030 = 18.6$ lb. of water must be evaporated per min. (1030 = average heat of vaporization of water under 26 to 28 in. of vacuum.) Now, 1 lb. of vapor at 102 to 125 deg. fahr. has an average volume of 270 cu. ft.; therefore, $18.6 \times 270 = 5022$ cu. ft. of vapor must be exhausted per min. to increase the vacuum from 26 to 28 in., which, while not impossible is manifestly impracticable for so small a condenser.

218. Classification of Condensers. — Steam condensers may be grouped into two broad classes:

Jet Condensers in which the steam and cooling water mingle and the steam is condensed by direct contact.

Surface Condensers in which the steam and cooling medium are in separate chambers and the heat is abstracted from the steam by conduction.

Jet condensers may be arranged with either **parallel flow,** in which the condensed steam, cooling water, and non-condensable gases flow in the same direction, and **counter current** in which the cooling water and steam with its air entrainment flow against each other. Jet condensers may also be classified as **low-level** and **barometric.** In the former the condensate and cooling water are removed and discharged against atmospheric pressure by means of the pump, and in the latter, withdrawal of the condensate and cooling water is effected by a pipe (34 ft. in length or more) called a **tail pipe,** or barometric column. With the low-level type the injection or condensing water is lifted into the condenser from the coldwell by the vacuum, while, with the barometric type, an injection or circulating pump is necessary to overcome part of the lift to the condensing chamber. Jet condensers in which the cooling water, condensate vapor, and non-condensable gases are withdrawn by a single piston pump are frequently designated as **low-vacuum** condensers because of the limited air capacity of the pump. In case the air-vapor mixture is removed by a separate pump or ejector, higher vacua are obtainable and the condenser is known as a **high-vacuum** jet condenser. Under certain conditions the air vapor and water can be removed by the kinetic action of the steam, in which case the condenser is designated as a **siphon** or **ejector** condenser.

Surface condensers may be classified, according to the nature of the cooling medium, as **water-cooled, air-cooled** or **evaporative.** In the latter type, the condensation of the steam is brought about by the evaporation of a fine spray or stream of water flowing across the surface of the tubes. Surface condensers may also be arranged, according to the relative position of the steam and water, as **standard,** in which the steam surrounds the tubes, and **water works,** in which the steam is inside the tubes.

219. Low-level Jet Condensers. — Figure 336 shows a section through the condensing chamber and water end of a **low-level low-vacuum** condenser illustrating the parallel-flow principle. This particular design is suitable for condensing small quantities of steam (25,000 lb. per hr. or less) where vacua over 26 in. are not necessary and where low first cost is of prime consideration. Unless the heat of the exhaust steam from the pump cylinder is used for feedwater heating or other purposes, the amount of steam required to operate the condenser may be prohibitive because of the extremely high water rate of the direct-acting type of pump. Operation is as follows:

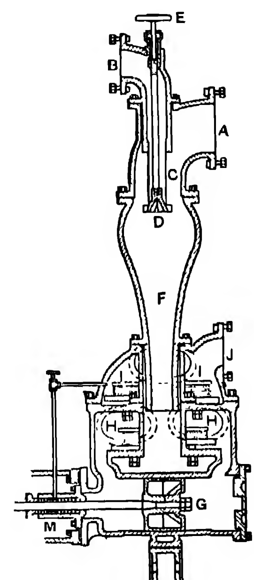


FIG. 336. Low-level Low-vacuum Jet Condenser with Direct-acting Steam-driven Vacuum Pump.

When the pump is started, a partial vacuum is created in the suction chamber above the valves *H*, *H*, in the cone *F*. As soon as sufficient air has been exhausted, cooling water enters at *B* with a velocity depending upon the degree of vacuum in chamber *F* and the suction head, and is divided into a fine spray by the adjustable serrated cone *D*. The spray mingles with the exhaust steam entering at *A*, and both move downward with diverse velocities. The steam gives up its heat to the water and condenses. The velocity of the steam diminishes in its downward path to zero, while the velocity of the water increases according to the laws of falling bodies. The condensed steam, cooling water, and air collect at the lower part of the condenser and are exhausted by the wet-air pump *G*, from which they are forced through opening *J* to the hotwell. The vacuum in chamber *F* will depend upon the vapor tension of the warm water in the bottom of the well, the amount of air carried along by the cooling water and steam, and the tightness of valves and joints. In case the water accumulates in the condenser cone *F*, either by reason of an increased supply or by a sluggishness or even stoppage of the pump, the

condensing surface is reduced to a minimum, as soon as the level of the water reaches the spray pipe and the spray becomes submerged, and only a small annular surface of water is exposed to the exhaust steam. The vacuum is immediately broken, and the exhaust steam escapes by blowing through the injection pipe and through the valves of the pump and out the discharge pipe at J, forcing the water ahead of it; consequently, flooding of the steam cylinder cannot occur. In starting up the condenser, a partial vacuum for inducing a flow of injection water into the condenser chamber may be created by the pump if the suction lift is not too great. Many engineers, however, prefer to install a small forced injection or priming pipe, the function of which is to condense sufficient steam to produce the necessary partial vacuum. This type can be used only where the injection nozzle is less than 18 to 20 ft. above the water supply.

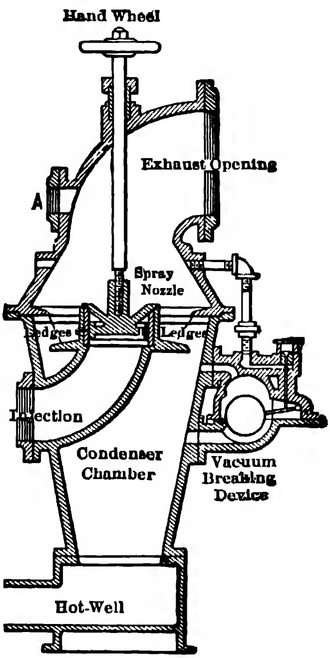


FIG. 337. Section through Condensing Chamber of a Low-level, Low-vacuum Jet Condenser with Vacuum Breaking Device.

Figure 337 shows a section through the condensing chamber of a vertical low-level jet condenser with an automatic vacuum-breaking device. The vacuum pump is either of the direct-acting or flywheel type, the latter being more economical in the use of steam. The injection water enters at opening marked "injection" and flows through the adjustable "spray" nozzle in a fine spray at an angle of about 45 degrees, and impinges on the conical sides of the upper condenser chamber. The spray falls from the sides to the projecting ledges shown in the illustration. The ledges prevent the spray from falling directly to the bottom of the chamber, and insure an efficient mingling of steam and cooling water. A perforated copper plate is substituted for the shelves when the force of the injection water is not sufficient to produce spray. The circulating water and condensed steam, together with the non-condensable gases, are drawn off at the bottom of the chamber. The vacuum-breaking device is shown at the right of the figure. When the rising water reaches the level of the float chamber, as in the case of an accidental stoppage of the air pumps, the float is raised and forces a check valve from its seat and allows an

inrush of air to break the vacuum, thus preventing further suction of water into the condenser and consequent flooding of the engine. A is the forced injection or "priming" inlet used in starting up when the suction lift is considerable.

Jet condensers of the type shown in Figs. 336 and 337 are not common in modern practice, because of the limited capacity of the piston type of vacuum pump. In present day practice the pump is of the centrifugal or rotary type and the air and water are usually withdrawn separately.

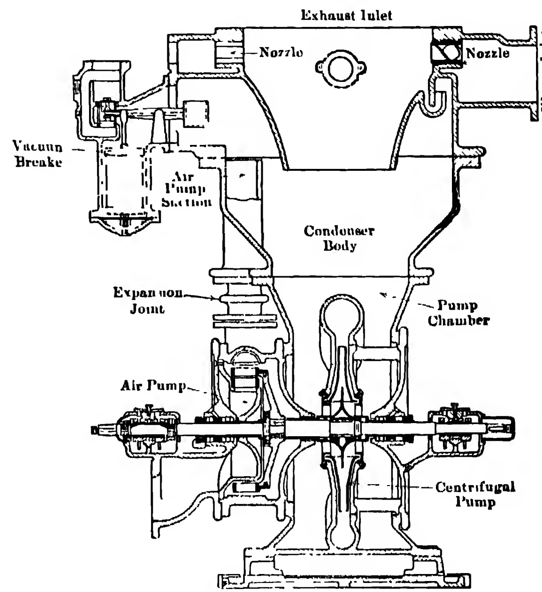


FIG. 338. Westinghouse-Leblanc Multi-jet High-vacuum Condenser.

The low-vacuum type of jet condenser is not suitable for high vacua because of the limited air capacity of the combined air and circulating water pump. Even with a tight system, considerable air is carried into the condenser with the circulating water, and efficient removal of the air necessitates a larger pump capacity than is usually furnished with this type of condenser. Low-level jet condensers may be operated with a high degree of vacuum by equipping them with independent air and circulating pumps. Examples of this type of jet condenser are illustrated in Figs. 338 and 340. Referring to Fig. 338, which gives a sectional view of the Leblanc type of condenser, steam enters the condensing chamber as indicated and meets the cooling water injected through the spray nozzles. The condensed steam and injection water fall to the bottom of the vessel and are removed by the centrifugal pump. The air-vapor mixture is withdrawn from near the top of the condenser body where the

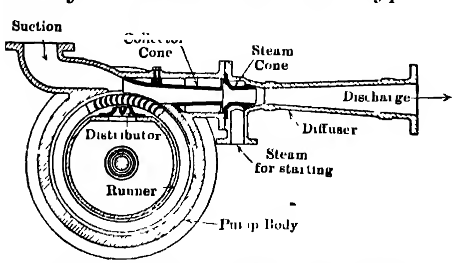


FIG. 339. Section through a Westinghouse-Leblanc Air Pump.

temperature is the lowest, into the suction inlet of the air pump. Referring to Fig. 339 which shows a section through the air pump, it will be seen that this device consists primarily of a multi-vane wheel in conjunction with an ejector. Sealing water is introduced into the central chamber, from which it is discharged through the "distributor." It is then caught up by the blades of the wheel, which is rotated at a suitable speed, and ejected into the discharge cone in the form of thin sheets having a high velocity. These sheets of water meet the sides of the "collector cone" and thus form a series of water pistons, each of which entraps a small pocket of air and forces it out against the atmospheric pressure. In passing through the air pump, the sealing water receives practically no increase in temperature, hence the same water may be used over and over again. The air pump rotor and main pump runner are mounted on the same shaft. In starting up, the condenser steam is turned into an auxiliary nozzle, Fig. 339, for a few moments, thus creating sufficient vacuum to start the regular flow of water through the air pump.

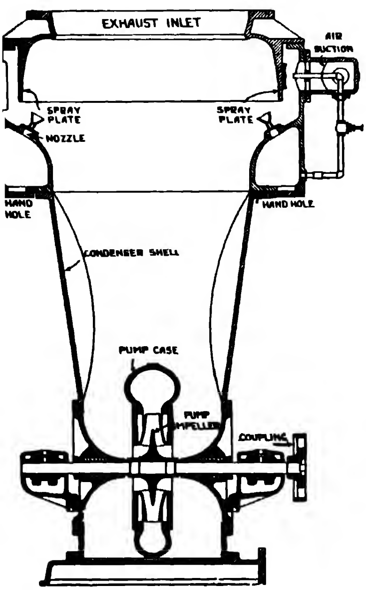


FIG. 340. C. H. Wheeler Low-leve Jet Condenser.

Low-level jet condensers equipped with centrifugal vacuum pumps are also known as **centrifugal jet condensers.** They may be either of the parallel-flow type, Fig. 338, with exhaust inlet at the top, or of the countercurrent type, Fig. 341, where the exhaust steam enters at the lower part of the condenser. Cooling water may be drawn into the chamber through orifices, Fig. 338, thereby producing a **spray,** or it may be distributed through the chamber by a series of pans or trays which break it up into a number of small streams and create a **rain** effect, Fig. 341.

The air-vapor mixture is removed from the condensing chamber by dry vacuum pumps of the piston or rotative type, hydraulic or "hurling-water" vacuum pumps, and steam ejectors. These auxiliaries are fully described in paragraphs (280) to (286) and need not be considered here. For maximum capacity the temperature of the air-vapor mixture should be lowered to practically that of the injection water. This is automatically effected in the counterflow type because the air is forced to pass

through the coldest part of the circulating water in its passage to the air pump. The same result may be obtained in the parallel-flow type by a proper location of the air-pump suction opening. The parallel-flow prin-

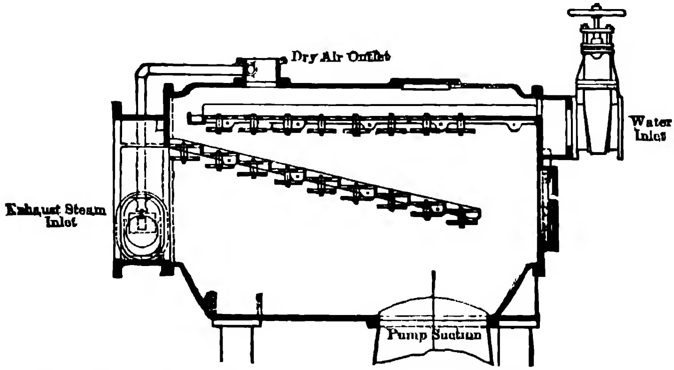


FIG. 341. Rain-type, Low-level, Centrifugal Jet Condenser.

ciple has the advantage over the counterflow in that the kinetic action of the steam forces part of the air-vapor mixture into the suction of the water pump and thereby reduces the quantity to be handled by the air pump.

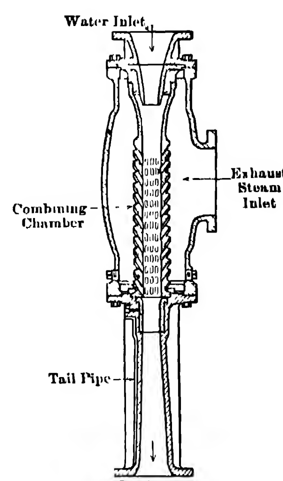


FIG. 342. Schutte "Eductor" Ejector Condenser.

The circulating water, condensate, and air-vapor mixture may be discharged without the aid of a barometric column or vacuum pump, by a special design of the circulating water and steam nozzles. Such a device is illustrated in Fig. 342 and is generally known as an **ejector condenser.** Referring to Fig. 342, it will be seen that the circulating water passes through a Venturi-shaped conduit, the central body of which is provided with a number of inclined nozzles for entrance of the exhaust steam. The shape of the water inlet is such as to convert the static pressure to velocity. The high-velocity water jet meets the exhaust entering through the circumferentially placed steam nozzle and forces the condensate with its air-vapor entrainment into the *tail pipe* or lower end of the conduit. The gradually increasing diameter of the latter is for the purpose of converting the velocity to pressure in order to overcome the resistance of the atmosphere. The tail pipe is always filled with water to prevent the air from entering the body of the condenser. This design

of condenser is low in first cost and in cost of operation, but is limited to comparatively small-sized units.

The condenser should be installed vertically, with 3 ft. of pipe between the strainer and the head of the condenser, and should be arranged as shown in Fig. 343. There should be a clear discharge of not less than 2 ft. below the bottom flange of the apparatus to the level of the water in the discharge pump, or hotwell. It is advisable that the end of the discharge pipe be sealed under water, unless there is a horizontal discharge main and a trap to the water-seal at the bend immediately under the condenser. Except with a condenser of very large size, a difference of level of 20 ft. between supply and discharge will usually give the necessary pressure of water at the condenser with full allowance for friction losses. Any type of circulating pump may be used for supplying the injection water. These condensers are made in all sizes, conforming with exhaust-pipe diameters of 1 1/2 to 24 in. The same amount of cooling water is required as for jet condensing, and vacua of 26 to 27 in. have been obtained under favorable conditions.

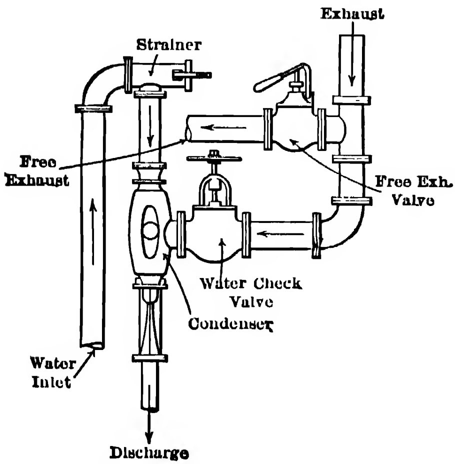


FIG. 343. Piping for Schutte Ejector Condenser.

The **multi-jet condenser** has been especially designed for condensing larger quantities of steam than the "Eductor Condenser" and maintaining a vacuum without the use of an air pump. The ratio of injection water per lb. of steam condensed is, for equal vacuum, considerably less than that required for the Eductor Condenser. This is due to the fact that in the multi-jet condenser the injection water is divided into a number of jets, the result of which is to bring the injection

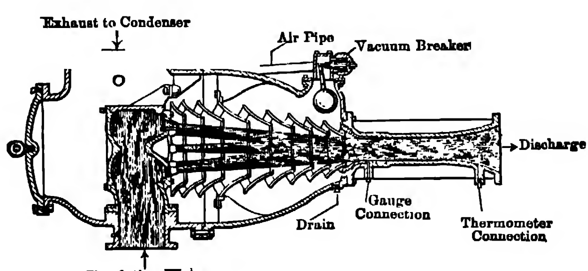


FIG. 344. Section through Condensing Chambers of Koerting Multi-jet Condenser.

water into more intimate contact with the steam. The steam flows into the main chamber through either side or top inlet, as desired.

The **Koerting Multi-jet condenser,** shown in Fig. 344, operates on the same principle as the Eductor Condenser, but has, instead of one central condensing jet, a number of converging jets, meeting and forming a single jet in the lower part of the combining tube. This tube is cast in one piece and consists of a series of concentric nozzles of gradually diminishing bore. The steam flows through the annular passages between the nozzles, which guide it so that it impinges at a suitable angle on the condensing jets. To insure satisfactory working under all conditions of load variation on turbines or engines to which these multi-jet condensers are attached, it is only necessary to supply the water to the condenser at a pressure, at the level of the water-inlet flanges, equal to a 21-ft. water column, or, say, 9 lb. per sq. in.

Modern multi-jet condensers are suitable for all sizes of prime movers up to 10,000-kw. capacity and in commercial operation are maintaining vacua of 28 to 29 in. referred to a 30-in. barometer with cooling water at 70 deg. fahr. This type of condenser requires more water than well designed low-level jet condensers, but the absence of vacuum pumps may offset this disadvantage.

TABLE 66

PERFORMANCE OF KOERTING LOW-LEVEL MULTI-JET CONDENSER*

	1	2	3	
Barometer, in. of mercury	29 22	29 32	29 41	29 37
Vacuum, in. of mercury	28 00	27 94	27.85	27.50
Absolute pressure, in. of mercury	1.22	1.38	1 66	1 87
Temperature injection water, deg. fahr.	77	78	79	76
Temperature of hotwell, deg. fahr.	80	84	87	88
Temperature difference, deg. fahr.	3	6	8	12
Vapor-tension corresponding to hotwell temperature, in. of mercury	1 032	1 174	1 292	1 334
Water pressure, lb. per sq. in.		8 0	8 5	12
Steam condensed, lb. per hr	10,900	20,000	26,800	41,800
Injection water, gal. per min.	7,600	7,000	7,000	7,300
Lb. injection water per lb. of steam	350	175	131	87.5

* Report of Prime Movers, N E.L.A., 13-22, 1922, p 23.

220. Barometric Condensers. — In the barometric type of jet condenser, the water and steam distribution system is substantially the same as in the low-level type, but the circulating water and condensate are removed from the condensing chamber by a barometric column or tail pipe of sufficient length to overcome the pressure of the atmosphere.

This necessitates locating the condensing chamber approximately 40 ft. above the end of the tail pipe. Water is lifted to the condenser chamber by any suitable type of pump, and, while the vacuum will raise the water to a considerable height, it is customary to allow only 18 ft. as the practical limit. The air-vapor mixture may be removed by any type of dry-air pump or ejector, or combination ejector and air pump. Figure 345 shows a section through an Ingersoll-Rand barometric condenser of the counter-flow type. The steam enters near the bottom of the condensing chamber, and the water at the top. The air-vapor entrainment is withdrawn at the top after it has been cooled to practically the temperature of the inlet water. Moisture carried over with the air is separated as indicated, and discharged into the hotwell by a small auxiliary tail pipe. This style of condenser is very simple in construction, having no moving parts to get out of order or constructed orifices to become clogged. There is no need for vacuum breaking devices and the steam can be blown directly through the column if necessary. One of the chief objections to this type of jet condenser is its extreme height.

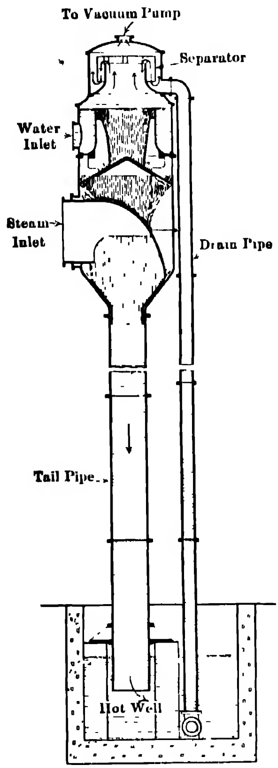


FIG. 345. Ingersoll-Rand Barometric Condenser. (Counter Current.)

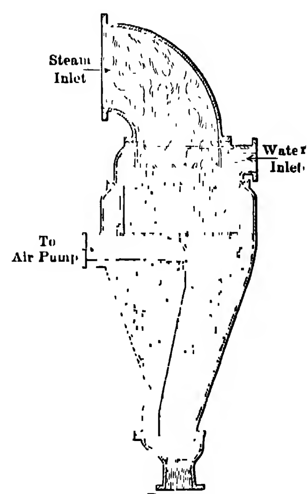


FIG. 346. Section through Condenser Head of a Wheeler Barometric Condenser.

The **barometric heater-condenser,** which differs in no way from the standard barometric-condenser design, has been used in power stations where the heat balance calls for this method of feedwater heating. The heater-condenser replaces the customary atmospheric heater and condenses the steam from the auxiliary steam drives and house turbine, utilizing the condensate from the main surface condenser as cooling water. See paragraph 265.

Figure 347 shows a section through the condensing head of a parallel-

flow barometric condenser with orifices and chamber arranged so that the kinetic action of the steam and circulating water will assist in discharging the condensate and air-vapor entrainment into the hotwell. No air pump is necessary for moderate vacua, since the velocity of the circulating water is sufficient to withdraw a limited quantity of air. For high vacua or where considerable air must be handled, a dry-air pump or ejector is necessary. Where the natural head of the circulating water is sufficient to overcome the difference between the total height of lift and head corresponding to vacuum, no circulating pump is necessary. Siphon condensers are capable of producing a high degree of vacuum when the amount of air-vapor entrainment is small, but as a general rule they are not recommended for vacua higher than 26 in. They are also limited to comparatively small sizes.

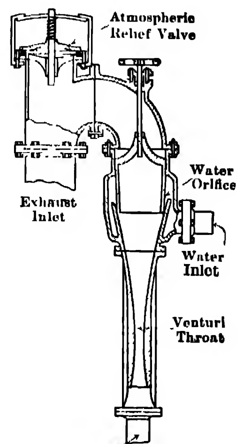


FIG. 347. Siphon Condenser.

221. Condensing Water: Jet Condensers. — In a jet condenser the cooling water and exhaust steam mingle, and the degree of vacuum is a function of the final or discharge temperature and the amount of non-condensable gases entrained with the steam and circulating water. The quantity of cooling water required depends upon its initial temperature, the temperature of the discharge water, and the total heat of the steam entering the condenser. If the steam in the low-pressure cylinder at exhaust is dry and saturated, and there is no air entrainment, the heat entering the condenser will correspond to the total heat of saturated steam at condenser pressure. This condition is not likely to occur in practice, since exhaust steam usually carries considerable moisture and there will be more or less air entrained with it. Furthermore, the cooling water contains air in varying amounts, so that the total amount of air entering the condenser may be considerable. Neglecting radiation and leakage, the heat absorbed by the cooling medium must be equal to that given up by the steam and its air entrainment. The heat exhange may be expressed

$$R = (H_m - q_2) \div (q_2 - q_o) \qquad (203)$$

in which

R = weight of injection water necessary to condense and cool 1 lb. of air-vapor mixture.

H_m = heat content of the air-vapor mixture at condenser pressure, B.t.u. per lb. above 32 deg. fahr.,

q_2 = heat of liquid of the discharge water, B.t.u. per lb.,
q_o = heat of liquid of the injection water, B.t.u. per lb.

In practice it is sufficiently accurate to neglect the influence of the air on the heat content of the exhaust steam and circulating water, and the mean specific heat of water under condenser conditions may be taken as unity, so that equation (203) may be written,

$$R = (H - t_2 + 32) \div (t_2 - t_o) \tag{204}$$

in which

H = heat content of the exhaust, B.t.u. per lb. above 32 deg. fahr.,
t_2 = temperature of the mixed condensate and discharge water, deg. fahr.,
t_o = temperature of the injection water, deg. fahr.

It has been shown (equation 146) that

$$H = H_i - H_r - A$$

in which

H_i = initial heat content of the steam entering the prime mover, B.t.u. per lb. above 32 deg. fahr.,
H_r = heat lost by radiation from the prime mover and exhaust piping, B.t.u. per lb. of steam admitted.
A = extraction for power, B.t.u. per lb. of steam admitted.

In a well-lagged piston engine with a short connection to the condenser, the loss to the surroundings, commonly called "radiation," varies from 0.3 to 2.0 per cent, but seldom exceeds 1 per cent of the total heat admitted, and in a turbine this loss is even less, and 0.2 per cent is a very liberal allowance. In view of the uncertainty of the numerical values of many of the factors entering into condenser calculations, it is sufficiently accurate for most purposes to ignore this small loss.

The temperature of the discharge water will approximate that of the vapor at its partial pressure. For air-free steam this will correspond to that of vapor at total condenser pressure. In high-vacuum jet condensers in which the air pressure is kept very low, the depression of the hotwell temperature will range from 6 to 10 degrees below that of vapor at condenser pressure, and in the ordinary low-vacuum condenser it may range from 15 to 25 degrees below. The shaded area in Fig. 348 shows the average range for a number of large installations. See also Fig. 474 for the estimated quantity of air to be removed from jet condensers.

Example 52. — Determine the amount of cooling water necessary per lb. of steam for a standard low-vacuum jet condenser operating under the

following conditions: Engine uses 16 lb. steam per i.hp-hr. initial pressure 140 lb. per sq. in. abs., superheat 50 deg. fahr., vacuum 4 in. Hg. abs., temperature of injection water 70 deg. fahr.

Solution. — From steam tables, $H_i = 1221$; assume $H_r = 1$ per cent of H_i, then

$$H = 1221 - 0.01\ (1221) - 2547/16 = 1050.$$

The temperature t_s of vapor corresponding to an absolute pressure of 4 in. = 126 deg. fahr.[1] Assume $t_2 = t_s - 15 = 111$.

Substituting these values in equation (204)

$$R = (1050 - 111 + 32) \div (111 - 70) = 23.7 \text{ lb.}$$

Neglecting the 1 per cent radiation loss, $R = 24.2$ lb. This small difference between the two calculated values of R shows the absurdity of including radiation loss for the assumed operating conditions.

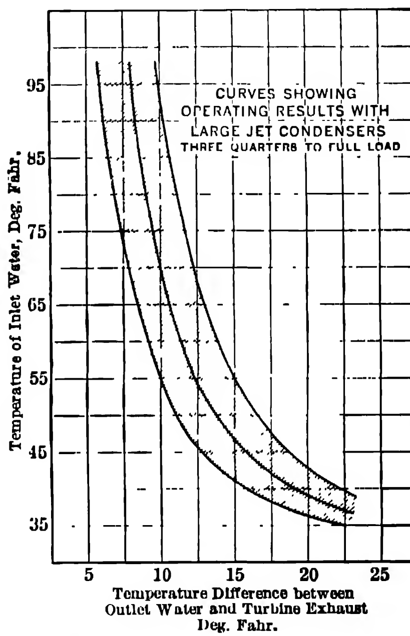


FIG. 348. Operating Results with Large Condensers.

Example 53. — Determine the amount of cooling water necessary per lb. of steam for a high-vacuum jet condenser operating under the following conditions: Turbine uses 13 lb. steam per kw-hr., initial pressure 165 lb. per sq. in. abs., superheat 120 deg. fahr., vacuum 2 in. Hg. abs., temperature of injection water 70 deg. fahr.

Solution. — From steam tables, $H_i = 1262$; $t_s = 92$; assume $e_1 = 0.95$; H_r is so small that it may well be neglected considering the many assumptions which must be made. Substituting these values in equation (146), noting that $W_1 = 13$, we have

$$H = 1262 - 3415 \div (13 \times 0.95) = 986.$$

Assume $t_2 = t_s - 5 = 92 - 5 = 87$.

[1] This is not the actual temperature in the condenser. The actual temperature will be that corresponding to the partial pressure of the vapor. For convenience in calculation the temperature in the condenser is assumed to correspond to that of the total pressure, and the temperature depression of the hotwell is then based on this hypothetical temperature. When the extent of air leakage and entrainment is known, the actual temperature in the condenser may be readily calculated.

Substitute $t_2 = 87$, $t_o = 70$, and $H = 986$ in equation (204) and solve, thus:

$$R = \frac{986 - 87 + 32}{87 - 70} = 54.8 \text{ lb.}$$

222. Water-cooled Surface Condensers. — With the exception of the water-works condenser, in which the weight of steam to be condensed is but a fraction of the weight of water passing, and in which the frictional resistances are to be kept very low, all water-cooled surface condensers are of the water-tube type; that is, the water is forced through the tubes. The latter vary in size from 5/8-in. outside diameter to 1 1/4-in. O.D. depending upon cleanliness of the circulating water and the design of condenser. They are composed of various alloys, usually **Muntz metal** (60 copper —40 zinc) and **ordinary brass** for reasonably pure fresh water, and **Admiralty brass** (70 copper —29 zinc —1 tin) for sea water and impure river waters. The tubes are generally held in tube sheets, composed of Muntz metal or Admiralty brass, by screwed brass ferrules and corset lace, fiber, or metallic packing, as shown in Fig. 351. In some designs the tubes are expanded into one tube sheet and packed in the other end, while in others they are expanded in both tube sheets. In the latter case, provision must be made for expansion due to change in temperature. The condenser shells are constructed in various shapes and are built of cast iron or steel. In the single-pass condenser there is but one chamber or water box between the tube sheets and condenser headers, while in the two-pass construction, the heads are divided into two compartments so that the circulating water will pass through one bank of tubes and return through the other. The exhaust from the engine or turbine enters the condenser shell at the top and circulates about the tubes down through the condenser. Condensate is withdrawn from the bottom of the shell and the air-vapor mixture from various points, as will be shown later.

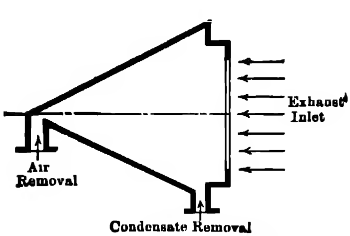


FIG. 349. Theoretically Correct Condenser Shape.

For high efficiency (1) steam should enter the condenser with the least practical resistance and the pressure drop through the condenser should be reduced to a minimum; (2) air should be rapidly cleared from the heat-transmitting surfaces, collected at suitable places, freed from entrained water, and removed at a low temperature with least expenditure of mechanical energy; (3) condensate should also be rapidly cleared from the heat-transmitting surfaces, freed from air and returned to the boilers at the maximum practical tempera-

ture; (4) circulating water should pass through the condenser with the least friction but at a velocity consistent with high efficiency. Figure 349 shows a theoretically correct condenser shape which embodies the prin-

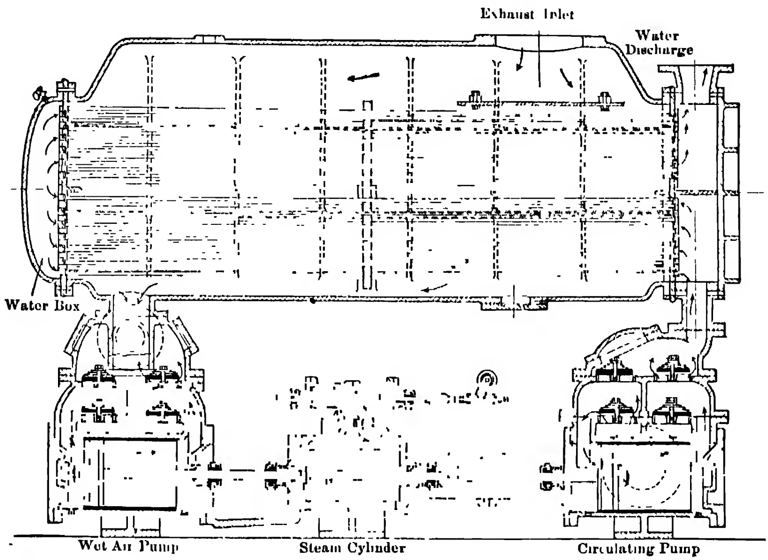


Fig. 350. Wheeler-Admiralty Surface Condenser Equipment (Piston-type Auxiliary Pumps).

ciples previously enumerated. It will be noted that the tube surface exposed to the action of the steam is a maximum at the exhaust inlet and decreases as the volume of steam diminishes, due to condensation. The air-vapor mixture flows directly to the discharge point at high velocity with minimum pressure drop, and the condensate gravitates from each row of tubes directly to the hotwell without blanketing the succeeding rows. Some of the means adopted in practice for simulating the action of the perfect condenser are best illustrated by a few descriptive examples.

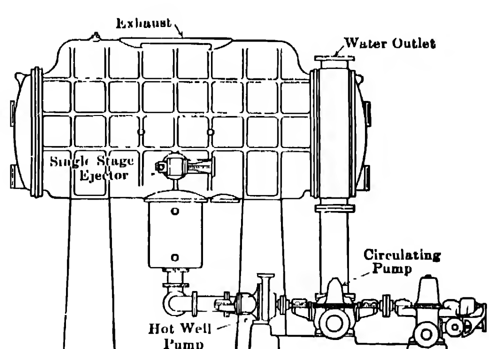


Fig. 351. Wheeler Admiralty Condenser Equipment (Centrifugal-type Auxiliary Pumps).

Figure 350 shows a section of a **Wheeler Admiralty** surface-condenser

equipment illustrating a well-known design which is intended for small engines or turbines and where vacua higher than 26 in. are not desired. The condenser is of the two-pass type and is mounted over a combined circulating and wet-air pump of the piston type. Since compactness and simplicity are of prime importance in this design and efficiency of little moment, no attempt is made to follow the principles of the " theoretically correct " shape. By installing independent circulating and hotwell pumps and by providing a suitable air ejector as shown in Fig. 351, the vacuum may be considerably improved.

Figure 352 shows the tube arrangement of the **Alberger Spiroflo** surface condenser showing a practical application of the basic principles illustrated in Fig. 349. The shell condenser heads and water boxes are of cast iron and the tube sheets are of brass. The tubes are expanded into one tube sheet, and in the other they are packed with a special fiber packing held in place by means of screw ferrule without lugs, thus providing for expansion or contraction. The condenser heads are divided into two compartments so that the circulating water will pass through one bank of tubes and return through the other. Steam enters through a deep, rectangular, steam-distributing dome, extending nearly the full length of the condenser shell. The sides of this distributing dome slope outward and become tangent to the main shell, leaving an arc of nearly 180 degrees of the tube surface exposed to the entering steam. The air-division plates, when taken in connection with the steam lanes and the grouping of the tubes, reduce the area of the path of steam in proportion to the volume flowing. The air-vapor mixture is drawn across a bank of tubes and reduced to practically the temperature of the inlet circulating water before being exhausted by the air pump. The circulating water flows countercurrently to the steam.

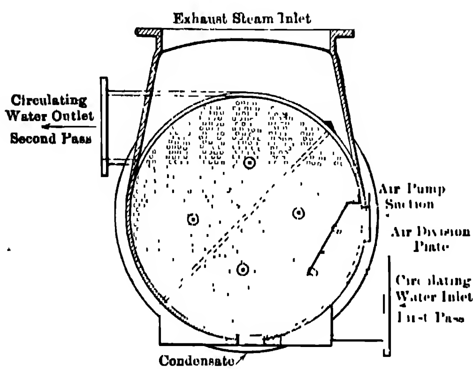


FIG. 352. Tube Layout — Alberger "Spiroflo" Condenser.

Figure 353 shows the tube arrangement for a 50,000 sq. ft. **Westinghouse** surface condenser which is of the radial flow type and fulfills the requirements of the correct principles outlined in Fig. 349. The exhaust steam, upon entering the condenser body under its condition of maximum specific volume, finds admission to the condensing surface at all points around the

circumference of the bank of tubes. The direction of flow of the gases is radially toward the center of the tube bank, at which point a connection is made with the air-removal apparatus. The condensate at the bottom of the condenser body is in contact with the exhaust steam. It is thus evident that the path of the steam flow is convergent and that the condensate is removed at high temperature, thereby meeting the conditions indicated by Fig. 349.

In the **C. H. Wheeler Duplex** condenser, Fig. 355, there is a clear passage for the steam from top to bottom, in addition to auxiliary side lanes, thereby insuring hot condensate and minimum pressure drop. The water passes are curves as indicated, so that the air-vapor mixture is sufficiently cooled before entering the air-pump suction.

This design gives, in effect, two hemispherical condensers with air suctions on each side, reduces the pneumatic resistance to a minimum, and also insures a uniform distribution of work in both upper and lower passes. With the high dome and central lane, the maximum surface is exposed to the incoming steam. The tubes in the upper pass are wider than those in the lower, permitting easy flow with minimum resistance to the lower pass.

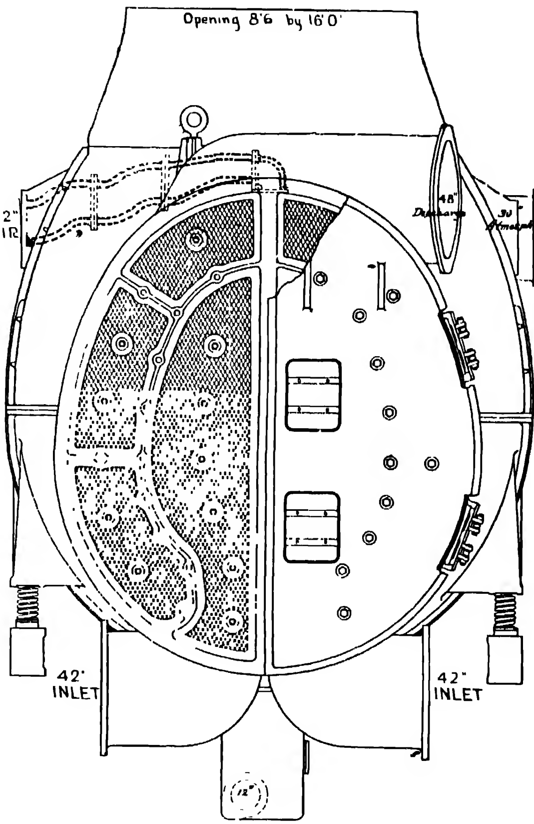


Fig. 353. Tube Sheet — 50,000 sq. ft. Westinghouse Surface Condenser.

In the **Ingersoll-Rand** condenser, Fig. 356, the tubes are arranged in stages, each stage consisting of a few rows on the same spacing. At the top the spacing is very wide both between the tubes and between rows.

The tubes in successive stages are arranged on smaller and smaller centers as indicated. The shell is heart-shaped and terminates in a narrow inlet. The circulating water is forced through the two upper groups in which the water makes a single pass to the discharge. Part of the circulating water is shunted to the coolers, flowing through these in parallel, and discharges into the bottom pass of the condenser. It then flows to the discharge where it joins the main body of water. Where two circulating pumps are used, the split boxes, as shown in Fig. 356, give control of water circulation at different velocities in different sections. Section "A" always works at high efficiency, even with very cold water and part loads. The lower sections, however, fall off in efficiency, and the water supply should be reduced in these independently of section "A."

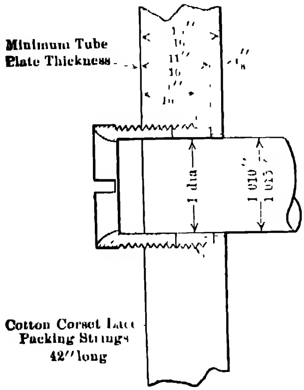


FIG. 354. Condenser Tube Packing.

Single-pass condensers are frequently installed where first cost is to be kept down and where high vacua are not essential; however, they are not necessarily inefficient, since (1) by the use of small diameter tubes any desired velocity may be obtained, (2) by proper distribution of the tube surface, blanketing may be reduced to a minimum, and (3) by the application of high-capacity air extractors results may be obtained comparable with any multi-pass arrangement. A notable installation of a high-vacuum single-pass surface-condenser is in the Saginaw River Steam Plant of the Consumers Power Company, Zilwaukee, Mich. (See *Power*, July 22, 1924, p. 122.)

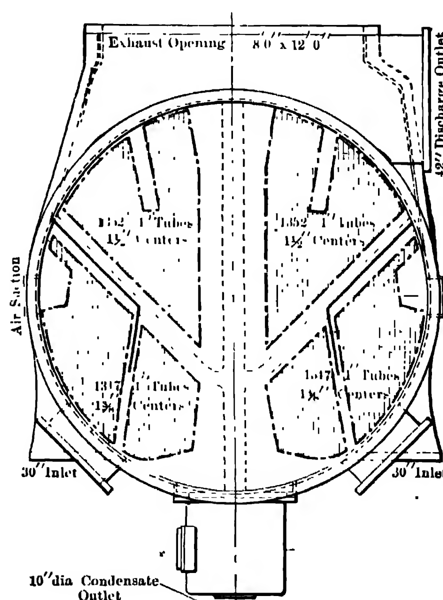


FIG. 355. Tube Sheet — C. H. Wheeler "Duplex" Condenser.

Surface Condenser Design: Trans. A.S.M.E., Vol. 43, 1921, p. 1059; Vol. 38, 1916, p. 672; Power Plant Engrg., May 15, 1924, p. 530.

Condenser Tube Packing: Report of Prime Movers Committee, N.E.L.A., 1923, Part A, p. 97.

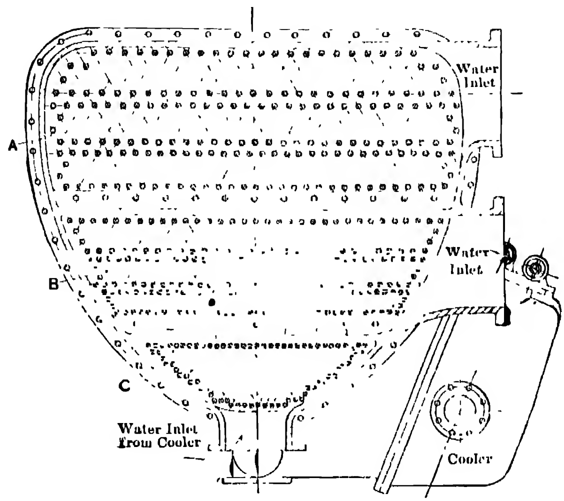


FIG. 356. Tube Sheet — Ingersoll-Rand Condenser.

223. Cooling Water: Surface Condensers. — Since the heat absorbed by the cooling water must equal that given up by the exhaust, neglecting radiation and leakage, the amount of cooling water may be determined as follows:

$$R = (H_m - q_1) \div (q_2 - q_o) \tag{205}$$

in which

q_1, q_2, and q_o = heat of liquid of the condensate, discharge and inlet water, respectively, B.t.u. per lb. above 32 deg. fahr.

Other notations as in equation (203).

Neglecting the heat content of the air entrainment and assuming a constant mean specific heat of unity for water, equation (205) may be written

$$R = (H - t_1 + 32) \div (t_2 - t_o) \tag{206}$$

in which

t_1 = temperature of the condensate, deg. fahr.

Other notations as in equation (204).

In the ordinary low-vacuum surface condenser, the depression of the hotwell temperature, t_1, below that corresponding to the total pressure in

the condenser, may range from 10 to 25 deg. fahr. depending upon the amount of air entrainment and the pressure drop through the condenser. An average figure is 15 deg. fahr. The temperature of the discharge water t_2 may also range from 15 to 30 deg. fahr. below that corresponding to the total pressure in the condenser. An average figure is 20.

For high-vacuum work the proper amount of circulating water is conservatively reached when 15 deg. fahr. is added to the temperature of the condensing water. This gives the proper velocity, at not excessive power, for good tube efficiency. Some conditions might warrant reducing this to 10 deg., which would require 50 per cent more water at approximately 240 per cent additional pumping power.

The following empirical rule for determining the terminal difference between the temperature of the steam corresponding to the vacuum in the condenser and that of the circulating water discharge gives results agreeing substantially with average surface-condenser practice

$$t_d = t - t_o \tag{206a}$$

in which

t_d = terminal difference, deg. fahr,
t = temperature corresponding to saturated vapor pressure $(p_o + B)$,
t_o = initial temperature of the circulating water, deg. fahr.,
p_o = pressure of saturated vapor corresponding to temperature t_o, in. of mercury,
B = coefficient, as follows:

VALUE OF COEFFICIENT B

Vacuum, In	B	Vacuum, In	B	Vacuum, In	B
1 00	0 20	1 75	0 35	3 00	0 50
1 25	0 25	2 00	0 40	3 50	0.60
1.50	0 30	2 50	0 45	4 00	0 70

Thus for $t_o = 70$ and a 2-in. vacuum: $p_o = 0.739$, $B = 0.40$, t corresponding to $0.739 + 0.40$ $(= 1.139) = 83.0$ deg. fahr., whence $t_d = 83 - 70 = 13$ deg. fahr.

Example 54. — (Low-vacuum condenser.) Required the weight of cooling water necessary to cool and condense 1 lb. of steam under the following conditions: Engine uses 16 lb. steam per indicated hp-hr., initial pressure 140 lb. per sq. in. abs., quality 0.99, initial temperature of the cooling water 70 deg. fahr., vacuum 4 in. Hg. abs.

Solution. — From the Mollier diagram or by calculation from steam tables, $H_i = 1184$ (approx.), $t_s = 126$, neglecting radiation loss $H_r = 0$.

From equation (146)

$$H = 1184 - 2547/16 = 1025.$$

Assume

$$t_1 = t_s - 15 = 111,\ t_2 = t_s - 20 = 106$$
$$R = (1025 - 111 + 32) \div (106 - 70) = 26.3 \text{ lb.}$$

With the modern high-vacuum surface condenser in connection with a practically air-tight system, the temperature of the condensate will be from 0 to 5 degrees lower than that corresponding to saturated vapor at condenser pressure, and the temperature of the discharge water will range from 8 to 15 degrees below that corresponding to the vacuum. The pressure drop through the condenser from exhaust inlet to air-pump suction varies with the type and size of condenser and the rate of driving, and ranges from 0.02 to 0.2 in. with an average at rated load of approximately 0.1 in.

Example 55. — (High-vacuum surface condenser.) Required the weight of cooling water necessary to cool and condense 1 lb. of steam under the following conditions: Turbine uses 12 lb. steam per kw-hr., initial pressure 200 lb. per sq. in. abs., superheat 150 deg. fahr., initial temperature of cooling water 70 deg. fahr., vacuum 1.5 in. Hg. abs.

Solution. — From steam tables, $H_i = 1283$. Assume $e = 0.95$ and neglect H_r.

Substituting these values in equation (146)

$$H = 1283 - 3415 \div (12 \times 0.95) = 983.$$

The corresponding temperature of vapor at 1.5 in. abs., $t_s = 91.7$ deg. fahr.

Assume $t_1 = t_s - 3 = 88.7$. From equation (206*a*), $t_s - t_2 = 10.1$.

Therefore $t_2 = 91.7 - 10.1 = 81.6$

$$\text{Whence } R = \frac{983 - 88.7 + 32}{81.6 - 70} = 79.8 \text{ lb.}$$

224. Heat Transmission through Condenser Tubes. — Numerous investigations have been conducted on special laboratory apparatus and on condensers in actual service for determining the heat transmission through condenser tubes, but the laws based on these results have been far from harmonious. In steam engine practice where the vacua are comparatively low, extreme refinement in design is unnecessary and simple empirical formulas for estimating the extent of cooling surface are sufficiently accurate. In modern high-vacuum practice, however, particularly for large turbo-generators where a fraction of an inch of change in vacuum greatly affects the economy of the prime mover, and where thousands of square feet of cooling surface are involved in a single unit, the older empirical rules are apt to lead to serious error. Despite the tremendous advance

in condenser design during the past few years, the art is still largely a matter of experience and the best rules are subject to arbitrary assumptions.

In any type of surface condenser, neglecting radiation and leakage, the heat absorbed by the cooling water, SUd, must be equal to that given up by the exhaust, $w_m\ (H_m - q_1)$, or

$$SUd = w_m\ (H_m - q_1) \tag{207}$$

in which

S = extent of cooling surface, sq. ft.,
U = experimentally determined mean coefficient of heat transmission, B.t.u. per hr., per deg. fahr. difference in temperature, d, per sq. ft.,
d = mean temperature difference between the steam and the circulating water, deg. fahr.,
w_m = weight of condensate plus the air entrainment, lb. per hr.,
H_m = heat content of the exhaust steam, moisture and air entrainment, B.t.u. per lb. above 32 deg. fahr.,
q_1 = heat of liquid of the condensate.

From equation (207) $$S = w_m\ (H_m - q_1) \div Ud \tag{208}$$

In view of the liberal factor allowed in estimating the value of U, and because of the uncertainty of the true value of d the influence of the heat content of the air entrainment becomes negligible and the equation may be written:

$$S = w\ (H - t_1 + 32) \div Ud \tag{209}$$

in which

w = weight of condensate, lb. per hr.,
H = heat content of the exhaust steam, B.t.u. above 32 deg. fahr.,
t_1 = temperature of the condensate, deg. fahr.

The coefficient U, as used in equations 207–210, refers to the *mean* or average value for the entire surface and not the *actual* value, since the latter varies widely for different parts of the condenser. The actual value varies from more than 1000 for air-free vapor in the first few rows of tubes (where the steam comes directly into contact with the cooling surface), to less than 50 in the bottom row (where the tubes may be practically blanketed with the condensed steam), and to 3 or less for tubes surrounded only by air. Tests by various investigators show that the actual value of U for a given temperature difference varies with

(*a*) material, thickness, size, shape, and cleanliness of the tubes;
(*b*) velocity of water through the tubes;

(*c*) percentage of air on the steam side of the tubes;
(*d*) critical velocity of the water in the tubes;
(*e*) extent of water blanketing on the steam side of the tubes;
(*f*) viscosity of the circulating water.

Taking the material coefficient, m, of plain copper tubes as 1.00, under similar conditions the heat transfer for other materials is approximately as follows: Admiralty brass, 0.98, Muntz metal 0.95, tin 0.79, Admiralty lead line 0.79, Monel metal 0.74, and Shelby steel 0.63 Corrosion, oxidation and pitting have a marked effect in reducing the heat transference and may lower the conductivity as much as 50 per cent. (See Fig. 357.)

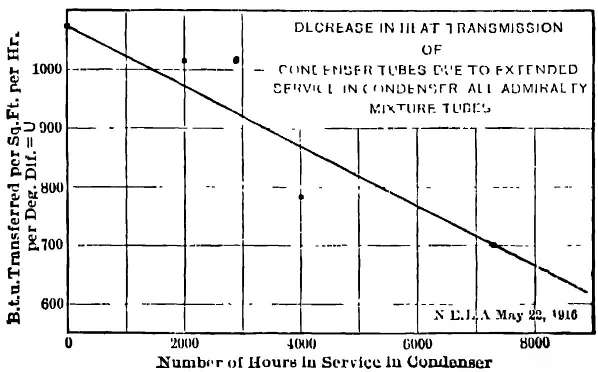


Fig. 357.

The cleanliness coefficient, c, is about 0.9 for such waters as those of New York or Chicago. The coefficient of heat transfer appears to decrease with the increase in diameter, but since the 1-in. tube, No. 18 B. W. G., is the most common in use this factor need not be considered for any other size.

Cleaning Surface Condenser Tubes: Elec. Jour., July, 1921, p. 313. Elec. World, Aug. 6, 1921, p. 273

The influence of the velocity of the water through the tubes on the coefficient of heat transfer is illustrated in Fig. 358 and Fig. 362. According to Orrok, the value of U, other conditions remaining constant, varies approximately as the square root or six-tenths power of the velocity. For the ordinary low-vacuum condenser the velocity through 1-in. standard tubes seldom exceeds 3 ft. per sec., whereas velocities as high as 8 ft. per sec. are not uncommon in the high-vacuum type. An average value for the latter is 6 ft. per sec. Except for a very low rate of flow (below that in average condenser practice), critical velocities need not be considered.

The effect of air on the heat transference is very marked, as is shown in Figs. 360–62. The depression of the hotwell temperature below that corresponding to the vacuum may be reduced by good design. Certain designs of dry-tube condensers may give hotwell temperatures somewhat higher than the average temperature in the condenser, and tests have been reported on several other designs in which the depression was zero. Orrok's investigations show that air entrainment reduces the heat transference approximately according to the law $(p_v \div p_c)^2$, in which p_v = pressure of the vapor and p_c the total pressure in the condenser.

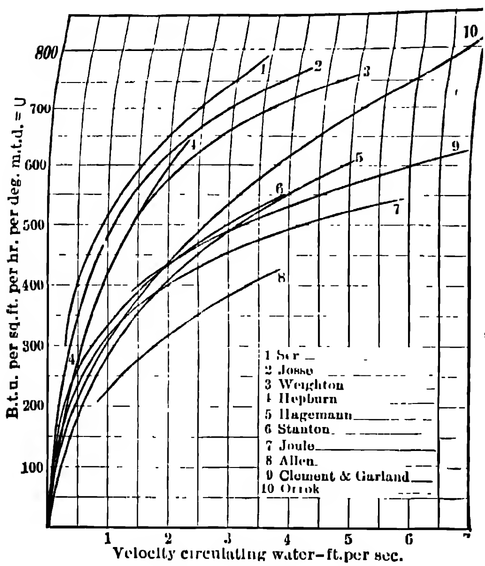


FIG. 358. Variation of Heat Transmission with Water Velocity.

The value of $(p_v \div p_c)^2$ varies within wide limits, but for tight condensers with efficient air pumps it may be taken as 0.95.

The Electrical Method of Detecting Surface Condenser Leakage: Power, Aug. 9, 1921, p. 217; Jan. 24, 1922, p. 126.

Air in Boiler Feedwater and Condenser: Power, March 1, 1921, p. 358.

How External Air Cooling Increases the Effectiveness of Condenser-tube Surface: Power, May 13, 1924, p 769

The reduction in heat transmission due to the thickness of water film on both sides of the tubes has been expressed mathematically, but it is customary in condenser design to include this factor in the assumed value of U.[1]

The coefficient of heat transmission increases with the mean temperature of the circulating water; that is, the warmer the water and the lower the vacuum, the smaller will be the mean temperature head required to transmit a practically constant amount of heat through the surface.

According to Orrok,

$$U = kcpmv^{0.6}/d^{\frac{1}{4}} \qquad (210)$$

[1] Trans. A.S.M.E., Vol. 35, 1915, p. 67; Indus. and Engrg. Chem. May 1924, p. 483.

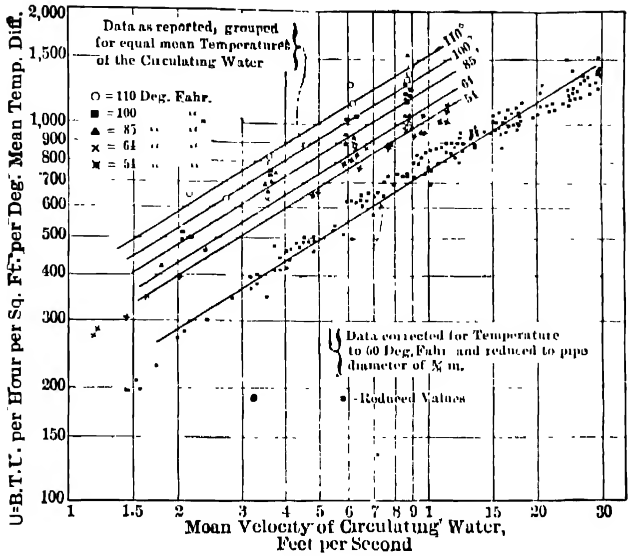


FIG. 359. Rate of Heat Transfer, Results of Tests by Geo. A. Orrok

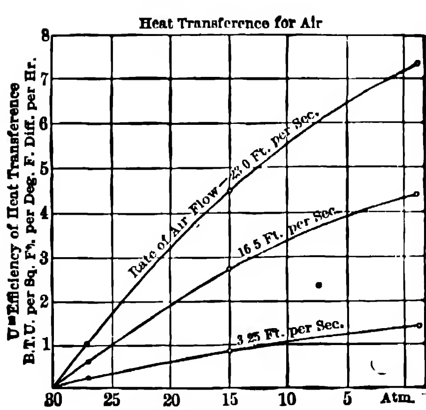


FIG. 360. Heat Transmission—Steam to Air.

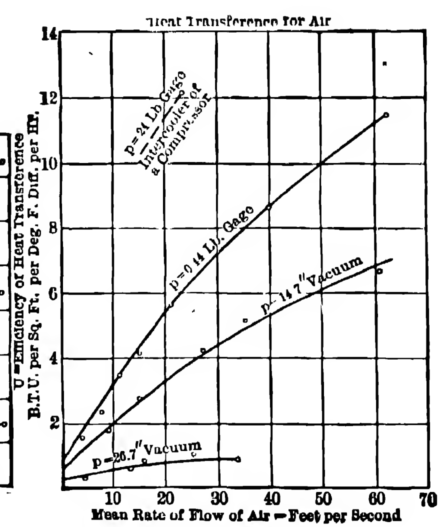


FIG. 361. Heat Transmission—Steam to Air.

in which

U = mean coefficient as previously defined and as used in connection with equations (208) and (209),
k = experimentally determined coefficient = 350 for average working conditions,
c = cleanliness coefficient,
p = air richness ratio = $(p_v \div p_c)^2$,
m = material coefficient,
v = velocity through the tube, ft. per sec.,
d = logarithmic mean temperature difference.

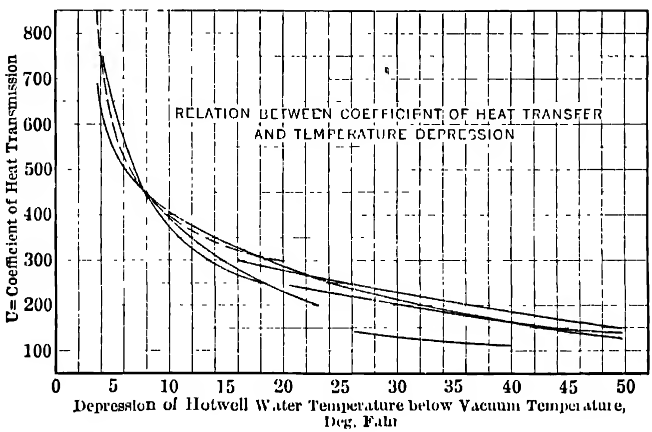


FIG. 362.

The following empirical rule, based on average good circulating water and clean condenser tubes, gives values of U which agree substantially with current practice in condenser design

$$U = 43.6 \sqrt[3]{t_o} \sqrt{v} = K \sqrt{v} \tag{211}$$

VALUE OF K FOR VARIOUS INITIAL TEMPERATURES OF CIRCULATING WATER

Initial Temp., Deg. Fahr.	K	Initial Temp., Deg. Fahr.	K	Initial Temp., Deg. Fahr.	K
40	145	60	170	80	187
45	155	65	174	90	196
50	160	70	180	100	202

Mean Temperature Difference. — It is definitely known that the quantity of heat passing through the cooling surface is proportional to some power

of the temperature difference at any instant; but the instantaneous temperature difference is indeterminate, consequently it is necessary to establish an average or mean temperature difference for the whole period of thermal contact of the steam and circulating water.

If t_s = temperature of the steam or hot substance,
t = any momentary temperature of the circulating water,
t_o = initial temperature of the circulating water,
t_2 = final temperature of the circulating water,
d = mean temperature difference,
Q = weight of circulating water, lb. per hr.,
S = extent of cooling surface, sq. ft.,
U_1 = instantaneous value of the coefficient of heat transfer,
U = mean coefficient of heat transfer for the entire period of heat exchange.

All temperatures deg. fahr.

Then the heat transmitted per hour through the elementary surface dS is $U_1\ (t_s - t)$.

Since the temperature rise for this period is dt, the heat absorbed by the circulating water per hour is Qdt (theoretically this should be $cQdt$ in

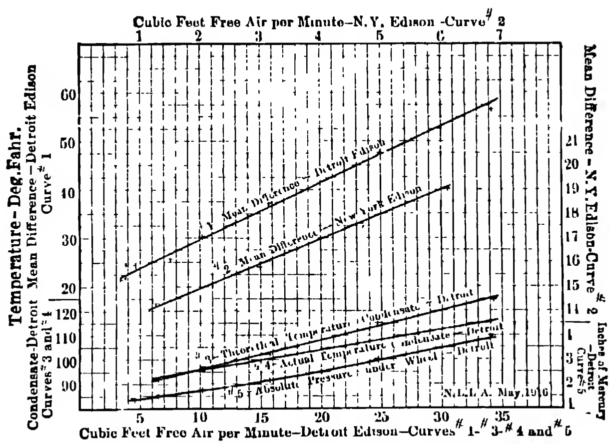


FIG. 363. Curves Showing Effect of Air Leakage on Condenser Efficiency.

which c is the mean specific heat of the water, but for all practical purposes the value of c may be taken as unity).

These two quantities must be equal, or

$$U_1\ (t_s - t)\ dS = Qdt, \tag{212}$$

from which

$$dS = Qdt \div U_1\ (t_s - t). \tag{213}$$

If the temperature of the steam is assumed to be constant, t_s is independent of t, and if the heat transmitted per hour is assumed to be directly proportional to temperature difference, U is likewise independent of t and $U_1 = U$; therefore the relation between rise in temperature of the circulating water and the surface traversed becomes

$$S = Q/U \times \int_{t_o}^{t_2} \frac{dt}{t_s - t} \tag{214}$$

$$= Q/U \times log_e\,[(t_s - t_o) \div (t_s - t_2)] \tag{215}$$

For the whole period of transfer,

$$SUd = Q\ (t_2 - t_o) \tag{216}$$

$$d = Q\ (t_2 - t_o) \div US \tag{217}$$

Combining equations (215) and (217) and reducing,

$$d = (t_2 - t_o) \div log_e\,[(t_s - t_o) \div (t_s - t_2)]. \tag{218}$$

This is known as the **logarithmic mean** temperature difference and is the one most commonly used in condenser design. The relation between temperature of the steam and that of the circulating water for this condition is shown by the solid line in Fig. 364.

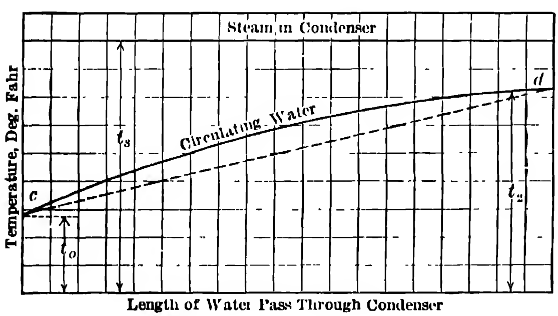


FIG. 364. Rise of Circulating-water Temperature in Condenser Tubes.

If the rise in temperature of the circulating water follows the dotted line cd, the mean temperature difference may be expressed

$$d = t_s - (t_2 + t_o) \div 2 \tag{219}$$

This is known as the **arithmetic mean** temperature difference and is used only for rough calculation or where other influencing factors can only be approximated. In general, when the temperature rise is approximately 20 deg. and where the difference between the discharge water temperature and steam temperature exceeds 15 deg. fahr., the arithmetic and logarithmic methods give substantially the same results, but where the difference is less than 15 deg. this is not true.

If the quantity of heat transmitted per hour is proportional to the nth power of the instantaneous temperature difference, as appears to be the

case in actual practice, and U is assumed to be constant

$$Qdt = U\ (t_s - t)^n\ dS \tag{220}$$

Integrating and reducing

$$S = \frac{Q}{U}\left[(t_s - t_o)^{1-n} - (t_s - t_2)^{1-n}\right]\frac{1}{1-n} \tag{221}$$

By assumption

$$SUd^n = Q\ (t_2 - t_o) \tag{222}$$

Therefore $$d = \left[\frac{(1-n)\ (t_2 - t_o)}{(t_s - t_o)^{1-n} - (t_s - t_n)^{1-n}}\right]^{\frac{1}{n}} \tag{223}$$

This is known as the *exponential mean* temperature difference. Orrok's[1] experiments lead to a value of $n = 0.875$. Loeb[2] assigns a value of $n = 0.9$. Because of the uncertainty of the value of U, it is sufficiently accurate for most purposes to take n = unity, which results in the logarithmic mean temperature difference.

According to equation (206)

$$R = (H - t_1 + 32) \div (t_2 - t_o)$$

And from equation (218)

$$d = (t_2 - t_o) \div log_e\left[(t_s - t_o) \div (t_s - t_2)\right]$$

Substituting these values of R and d in equation (209) and solving, we have, as a general rule for heat transmission in surface condensers,

$$S = \frac{wR}{U}\,log_e\,\frac{t_s - t_o}{t_s - t_2} \tag{224}$$

in which

w = weight of condensate, lb. per hr.,
R = ratio by weight of circulating water to condensate or the lb. of circulating water per lb. of condensate,
U = mean coefficient of heat transfer, B.t.u. per hr. per sq. ft. per logarithmic mean temperature difference,
t_s = temperature of the steam,
t_2 = temperature of the discharge water, deg. fahr.,
t_o = temperature of the intake water, deg. fahr.

Orrok[3] gives the following general rule for high-vacuum surface con

[1] Trans. A S.M.E., Nov., 1916.
[2] Jour. Am. Soc. Naval Engrs., Vol. 27, May, 1915.
[3] Trans. A.S.M.E., Vol 38, 1916.

densers operating under favorable conditions

$$S = \frac{Q}{40.3\, v^{0.6}} \left[(t_s - t_o)^{\frac{1}{4}} - (t_s - t_2)^{\frac{1}{4}} \right] \tag{225}$$

The number of tubes of given diameter and thickness which will pass a given quantity of water per hr. at a predetermined velocity may be calculated as follows:

Let d = outside diameter of the tube, in.,
n = number of tubes in each pass of the condenser,
l = length of water travel, or total tube length, ft.

Then, $S = \pi dnl/12$, whence $l = 3.83S \div dn$. (226)

By simple arithmetical calculation it may be shown that

$$n = Q \div 1233v(d - 2t)^2 \tag{227}$$

in which t = thickness of the tube, in.

Example 56. — (Low-vacuum condenser.) Approximate the amount of condenser cooling surface for a 1000-hp. compound engine operating under the following conditions: Water rate 16 lb. per i.hp-hr., initial steam pressure 140 lb. abs., initial quality 0.99, inlet of temperature of circulating water 70 deg. fahr., vacuum 4 in. Hg. abs.

Solution. — From steam tables, t_s corresponding to 4 in. of Hg. abs. = 126 deg. fahr. From equation (146), $H = 867.6 \times 0.99 + 324.6 - 2457/16 = 1025$ (approx.).

In the ordinary engine condenser, considerable air will be carried with the steam into the condenser, and the hotwell depression may range from 5 to 20 degrees; assume the depression to be 10 degrees, then $t_1 = t_s - 10 = 126 - 10 = 116$ deg. fahr.

Any value may be assumed for t_2 greater than t_o and less than t_1. The nearer t_2 is to t_o, the greater must be the quantity of circulating water per lb. of condensate. On the other hand, the nearer t_2 is to t_o, the less is the mean temperature difference d, and hence the greater must be the cooling surface for a given weight of condensate. When water is cheap and the head pumped against is small, t_2 may be given a lower value than when water is costly and the discharge head is large. In average engine-condenser practice, t_2 may range from 5 to 20 degrees below t_1; assume it to be 10 degrees, then $t_2 = t_1 - 10 = 116 - 10 = 106$ deg. fahr. Equation (206*a*) gives $t_2 = 105.5$.

Because of the great latitude in assuming values of t_1 and t_2, it is sufficiently accurate to use the arithmetical mean, or

$$d = 126 - (70 + 106)/2 = 38.0.$$

In engine practice a very liberal factor is allowed in assuming a value for U, because of the possible reduction in heat transmission caused by the deposit of cylinder oil on the tubes and because of excessive air entrainment. For the usual engine type of condenser with clean circulating water,

a safe value is $U = 300$. According to equation (211), $U = 300$ for $v = 2.8$ ft. per sec.

Substituting these values in equation (209) and reducing

$$S = 16{,}000\ (1025 - 116 + 32) \div 300 \times 38 = 1320 \text{ sq. ft.}$$

This corresponds to approximately 12.1 lb. of condensate per hr. per sq. ft. of tube surface. An average figure commonly quoted for engine condensers is 10 lb. of steam per hr. per sq. ft. of tube surface for 24 to 26-in. vacuum with 70-degree cooling water. A rough rule is to allow 2 sq. ft. of cooling surface per i.hp.

TABLE 67

TESTS OF 50,000 SQ. FT. SURFACE CONDENSERS

Index	1	2	3	4
Barometer, mercury at 58.1 deg. fahr.	29.93	30.07	30.00	30.38
Vacuum, mercury at 58.1 deg. fahr.	28.51	28.01	28.61	29.37
Absolute pressure, in. of Hg	1.32	2.03	1.39	1.01
Temperatures, deg. fahr.:				
Condensate	87.0	91.5	86.0	62.0
Inlet water	78.2	75.1	70.8	50.0
Outlet water	83.0	86.4	80.9	58.0
Rise in water	4.8	11.3	10.1	8.0
Corresponding to vacuum	91.6	100.6	89.4	65.3
Between condensate and steam	4.6	9.1	3.4	3.3
Mean difference (log)	10.8	19.4	12.9	9.41
Condensate, lb. per hr., thousands	180.7	373.7	357.0	300.0
Water, gal. per min., thousands	72.5	63.8	64.7	71.5
Lb. water per lb. of steam	200.0	98.5	91.0	119.0
Air leakage, cu. ft. per min.	9.9	6.5	16.9	...
Coefficient of heat transmission	322	372	190	598

1, 2. Unit #12, Seventy-fourth St. Station, Interborough Rapid Trans., 1916.
3. Unit #1, Fifty-ninth St. Station, Interborough Rapid Trans., 1920.
4. Unit #1, Williamsburg Station, Brooklyn Rapid Trans., 1920.

Example 57. — (High-vacuum condenser.) Calculate the amount of tube surface required for a 10,000-kw. turbine operating under the following conditions: Water rate 12.0 lb. per kw-hr., initial absolute pressure 200 lb. per sq. in., superheat 150 deg. fahr., temperature of circulating water 70 deg. fahr., vacuum 1.5 in. Hg. abs., water velocity through tubes 6 ft. per sec., cooling surface to consist of 1-in. (18 B.W.G.) Admiralty tubes.

Solution. — For maximum theoretical efficiency $t_2 = t_1 = t_s$. This condition is possible only for air-free vapor, perfect heat transmission, and no pressure drop between turbine nozzle and air-pump suction. (From steam tables, t_s corresponding to an absolute back pressure of 1.5 in. = 91.7.) Assume $t_1 = t_s - 3 = 91.7 - 3 = 88.7$. The temperature of the condensate varies from $t_1 = t_s - 0$ to $t_1 = t_s - 4$ deg. fahr., and t_2 varies from $t_2 = t_s - 4$ to $t_s - 12$ deg. fahr. From equation (206*a*) $t_s - t_2 = 10.1$.

From Example 55, the conditions of which are the same as for this example, we find $R = 79.8$; $t_2 = 81.6$.

The condenser must be designed for the maximum load when the circulating water is at its highest temperature, and where reasonably good water is not obtainable a suitable factor should be allowed for dirty, oxidized tubes and the presence of undue amounts of air. For this reason a much lower value of U is ordinarily assumed than is possible with everything in first-class shape. According to equation (211), $U = 440$ for a velocity of 6 ft. per sec.

TABLE 68

MODERN SURFACE—CONDENSER PROPORTIONS

Initial pressure 275 lb gage and under.
Initial temperature 600 deg fahr and under
No bleeding stages

Size of Turbo-generator	Tube Surface Sq Ft	Sq Ft Tube Surface per Kw.	Size of Turbo-generator	Tube Surface Sq Ft.	Sq. Ft. Tube Surface per K
500	1,500	3 00 3 50	10,000	17,500	1 75–2 25
1000	2,750	2 75–3 25	15,000	25,000	1 67–2 00
2000	5,000	2 50–3 00	20,000	32,000	1 60
5000	10,000	2 00–2 50	35,000	56,000	1 60
7500	13,500	1 80–2 25	40,000	60,000	1 50

TABLE 69

LARGE SURFACE—CONDENSER INSTALLATIONS

(1921–1924)

Station	Size of Turbo-generator, Kw.	Initial Pressure Lb Gage	Steam Temperature, Deg Fahr.	Condenser Surface, Sq Ft.	Sq Ft Surface per Kw. Rated Capacity
Cahokia	30,000	300	700	53,000	1.76
Barbados	25,000	300	625	45,000	1 77
Calumet	30,000	300	622	52,000	1 73
Colfax	60,000	265	611	100,000	1 66
*Gennevilliers	40,000	313	705	37,650	0 94
Hell Gate	40,000	250	607	57,000	1 43
Hudson Ave.	50,000	265	611	70,000	1 40
Kearney	35,000	325	700	50,000	1 43
Lansing, Mich	15,000	275	609	27,000	1 80
Marysville	30,000	275	700	30,350	1 02
Northeast, Kan	30,000	280	650	45,000	1 50
South Meadow	20,000	250	640	30,000	1 50
Springdale	25,000	300	690	32,000	1 28
Steel Point	10,000	200	500	16,250	1 62
Trenton Channel	50,000	400	700	52,000	1 04
Wabash River	20,000	300	650	40,000	2 00
Waukegan	25,000	350	700	32,000	1 60
Weymouth	30,000	375	700	45,000	1 50

* City of Paris.

Substituting these values in equation (224),

$$S = \frac{12 \times 10{,}000 \times 81.6}{440} \times \log_e \frac{91.7 - 70}{91.7 - 81.6}$$

$$= 22{,}250 \times 0.7655 = 17{,}000 \text{ sq. ft. (approx.)}.$$

corresponding to 1.7 sq. ft. per kw. of turbine rating.

See Tables 68 and 69 for modern condenser proportions.

Condensing Equipment: Report of Prime Movers Committee, N.E.L.A., Jan., 1926.
Determining the Economical Interval between Cleanings of Condenser Tubes. Power, Nov. 20, 1923, p. 803.

225. Dry-air Surface Condensers. — Ordinary atmospheric air may be used as a condensing and cooling medium for surface condensers, but the volume of air to be circulated and the extent of cooling surface necessary to effect the desired results are very high because of the low density and specific heat of the air and the poor heat transmission from steam to air. The power required to circulate the air is also very high. A few plants in Western Australia and in Central Africa were equipped with dry-air condensers in the early days when the internal combustion engine was undeveloped and long distance transmission lines were unknown, but they have long since been abandoned and no new plant of this type has been installed for years. The modern steam automobile is an example of the application of atmospheric air for condensing steam, but the quantities involved are comparatively small and no other cooling medium is available. Some idea of the enormous extent of cooling surface and the tremendous volumes of air necessary to cool even a small quantity of steam is evidenced by the performance of the old abandoned air-cooled plant in the City of Kalgoorlie, West Australia. The plant, rated at 2000 hp., had 45,000 sq. ft. of condenser surface and required 600,000 cu. ft. of air per min. at 80 deg. fahr. to condense the steam at rated load. The fans circulating the air required 200 hp. or 10 per cent of the station output for their operation. The vacuum ranged from 3.6 in. (referred to 30-in. barometer) with air at 108 deg. fahr. temperature to 22 in. with air at 43 deg. fahr.

The volume of air, under atmospheric conditions, necessary to condense steam to any given temperature may be determined as follows:

Let H = heat content of the steam at condenser pressure,
t_s = temperature of the vapor in the condenser,
t_1 = temperature of the condensed steam,
t = temperature of the air entering condenser,
t_o = temperature of the air leaving condenser,
V = volume of air in cu. ft. necessary to condense and cool 1 lb. of steam,

B = specific weight of air under atmospheric conditions,
C = mean specific heat of air under atmospheric conditions,
d = mean temperature difference between the air and steam,
S = cooling surface in sq. ft.,
U = coefficient of heat transmission, B.t.u. per sq. ft. per degree difference in temperature per hr.

Since the heat absorbed by the air must be equal to the heat given up by the steam, neglecting radiation, we have

$$VBC(t_o - t) = H - t_1 + 32, \tag{228}$$

from which

$$V = \frac{H - t_1 + 32}{BC(t_o - t)}. \tag{229}$$

For practical purposes, C may be taken as the specific heat of dry air, the error due to this assumption being negligible even if the air is saturated with moisture.

Example 58. — How many cu. ft. of air are necessary to condense and cool 1 lb. of saturated steam under the following conditions: Vacuum 10 in. Hg. abs., temperature of entering air, leaving air, and condensed steam, 60, 110, and 140 deg. fahr., respectively?

Solution. — Here $H = 1130$ (from steam tables),
$t_o = 110$, $t_1 = 140$, $t = 60$, $C = 0.24$, $B = 0.075$.

Substituting these values in equation (229),

$V = \dfrac{1130 - 140 + 32}{0.075 \times 0.24\ (110 - 60)} = 1135$ cu. ft. of air necessary to condense 1 lb. of steam under the given conditions.

The proper area of cooling surface depends upon the value of the coefficient of heat transmission, which varies with the velocity and humidity of the air and character of the cooling surface. Accurate data are not available on this point.

A few experiments made at the Armour Institute of Technology gave values of $U = 10$ to 25 B.t.u. per hr. per sq. ft. per deg. fahr. difference in temperature for air velocities of 500 to 4000 ft. per min. for corrugated-steel sheeting 1/8 in. thick. Assuming these values of U for the above example, $S = 1.5$ sq. ft. of cooling surface per lb. of steam condensed per hr. for air velocity of 4000 ft. per min., and $S = 3.7$ sq. ft. for a velocity of 500 ft. per min.

Air heaters of the bleeder type are identical in theory with the dry-air surface condenser, but the primary object in this case is the heating of the air and not the condensation of the steam. With the heating surface made

up of small tubes and extended sheet-metal fins, as in the **Griscom-Russell U-fin Preheater,** the coefficient of heat transmission is very high and large quantities of air may be heated in a comparatively small chamber.

226. Saturated-air Surface Condensers. — If, instead of using ordinary atmospheric air as a cooling medium, a small amount of water is permitted to trickle over the surface of the tubes, so that the air leaving the condenser is saturated or nearly so, the volume of air necessary to effect a given cooling effect is greatly reduced. This reduction in air volume is possible because the water is vaporized and absorbs a considerable portion of the heat given up by the steam in condensing. While the air itself absorbs part of the heat, its primary object in this connection is to carry away the vapor. Condensers of this type are not much in evidence, but a few installations are to be found in small plants where circulating water is scarce and vacua above 20 in. are not necessary. Figure 365 shows vertical and horizontal sections of a Pennel saturated-air surface condenser, illustrating the principles of this class of apparatus. The apparatus consists of an upright cylindrical shell containing a number of vertical 4-in. steel tubes through which air is drawn by natural draft. A centrifugal pump circulates about one-half gallon of water per hp. per min. from a cistern below the condenser. The water, flowing over the upper tube sheet and then descending the tubes by gravity, forms a film over their entire interior surface.

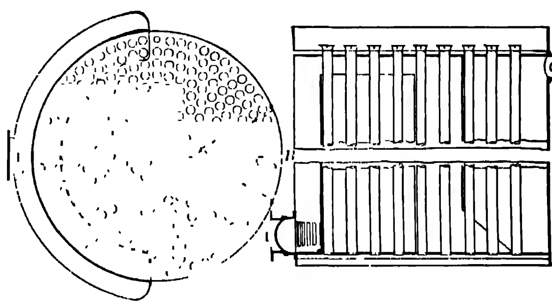


FIG. 365. Pennel Saturated-air Surface Condenser.

The condensing action is as follows: The current of exhaust steam entering the side of the sheet at A is caused by suitable baffle plates to circulate among the tubes, and in condensing gives up its latent heat to the water film, which wholly or partially evaporates, saturating the ascending current of air to its own temperature. The upward current of hot vapor-laden air carries off the heat into the atmosphere. The cooling water that is not evaporated and lost to the atmosphere falls into the cistern, being kept constant by a float governing a valve in the supply pipe. The non-condensable gases collect at C, where they are removed by the dry-air vacuum pump and discharged into the hotwell. An excel-

lent feature of this device is that the film of water on the cooling surface is secured without interference with the ascending air currents and also without the use of sprays through small orifices likely to become clogged with rust or sediment.

The weight of water evaporated in this type of condenser is approximately equal to that of the condensed steam, and the volume of air is about 2 per cent of that which would be required if dry air only were used. These volumes, of course, vary widely with the temperature and relative humidity of the air, and the heat to be abstracted per lb. of steam.

By forcing air through the tubes mechanically and by injecting a small amount of water spray into the entering air, the same effect may be produced as with natural draft, and much less cooling surface is required, but the work done in moving the air is greatly increased. Considerable experimental work has been carried out with this type of apparatus for situations where circulating water is costly, as, for example, in isolated stations for office buildings and the like; and while high vacua are readily obtained, the power requirements of the fan are excessive and the scale deposited on the cooling surface by the evaporation of the water necessitates frequent shutting down of the apparatus for cleaning purposes. The air and water requirements are substantially the same as for the natural draft apparatus, but it should be remembered that any proportion of air and water may be used, ranging from all water to all atmospheric air.

See Example 121 for calculations.

227. Evaporative Surface Condensers. — While saturated-air surface condensers are in reality evaporative condensers, the latter term is usually applied to those in which the medium to be condensed is on the inside of the cooling surface and the condensing water flows over the surface in the form of a thin film. This type of condenser is in common use for condensing ammonia in refrigerating plants but is not much in evidence in the modern steam power plant. When used for steam condensing purposes, the apparatus consists usually of a number of copper, brass, wrought- or cast-iron tubes arranged horizontally or vertically and connected to manifolds or chambers at each end. The exhaust steam passes through the tubes and a thin film of water is allowed to flow over the external surfaces. The cooling effect is brought about by the evaporation of part of the circulating water, and the general principle of operation is the same as that of the saturated-air condenser described above. Evaporation is sometimes hastened by constructing a flue over the tubes, thereby creating a natural draft, or by means of fans. With horizontal cast-iron tubes and natural draft, vacua from 23 to 27 in. are readily maintained with a cooling surface of approximately .8 sq. ft. per lb. of steam condensed per

hour. With vertical brass tubes and fan draft, 8 lb. of steam per hour per sq. ft. of cooling surface is not an unusual figure. The amount of cooling water evaporated per lb. of steam varies from .8 to 1 lb., depending upon the draft. The power necessary to operate the pumps and fans varies from 1 to 10 per cent of the total output of the plant. For an interesting discussion of evaporative condensers, the reader is referred to the admirable article by Oldham in the *Proceedings of the Institute of Mechanical Engineers*, 1899, and reproduced as a serial in *Engineering* (London), April 28 to June 30, 1899. The following test of a vertical cast-iron tube evaporative surface condenser (Table 70) will give some idea of the performance of this type of condenser. This condenser consists of two rows of 4-in. vertical cast-iron pipes connected at the top by *U* bends and at the bottom by cast-iron manifolds. A perforated iron trough distributes the water over the center of the bend and causes it to flow in a thin stream over the surface of the tubes. A wet-air pump is used for withdrawing the condensed steam and air. No fan is used for hastening evaporation.

See Chapter XXIV, for evaporative surface-condenser calculations.

Evaporative Condensers Engrg., Lond., May 5, 1889, pp. 432, 442, 447; Engrg., May 19, 1899, p. 661, June 2, 1899, p. 721, June 30, 1899, p. 861; Trans. A.S.M.E., 14–696; Power, Nov 16, 1909; Prac Engr. U. S., June, 1910, p. 346.

TABLE 70

TEST OF A CAST-IRON, VERTICAL TUBE, EVAPORATIVE SURFACE CONDENSER, NATURAL DRAFT

Date	Sept. 12	Sept. 13
Weather	Wet	Fine
Barometer	29.8	29.5
Temperature of air	?	60
Cooling surface, external	272	272
Duration of trial, min.	99	115
Weight of steam condensed, lb	800	800
Boiler pressure, lb. gage	60	60
Weight of water in circulation, lb	1830	1830
Weight of fresh water added, lb	600	640
Vacuum in condenser, in Hg	23.36	24.1
Initial temperature of circulating water, deg. fahr	117.5	113.9
Final temperature of circulating water, deg. fahr	128.4	125
Temperature of "makeup" water, deg. fahr	58	58
Temperature of water in hotwell, deg. fahr	136.5	131.8
Weight of steam condensed, lb. per hr.	485	427
Weight of water circulated, lb. per hr.	6786	?
Weight of "makeup" water added, lb. per hr.	364	334
Weight of steam condensed per lb. per sq. ft. of cooling surface per hr.	1.8	1.54
Weight of "makeup" water per lb. of steam condensed, lb.	0.75	0.80

228. Location and Arrangement of Condenser and Auxiliaries. — In the modern steam plant one sees two general arrangements of condensers and auxiliaries: *(1) the* **independent** *or subdivided system, in which each engine or turbine is provided with its own condenser, air and circulating pumps, and (2) the* **central** *system, in which the condensers and auxiliaries* are grouped together. In the latter system one condenser ordinarily suffices for all engines.

Independent System. — This system is used in practically all electric power stations of whatever size. The condenser should be piped as directly as possible to the engine or turbine exhaust opening, in order to avoid excessive pressure drop. If possible, the condenser should be placed below the prime mover so that all condensation may gravitate into it. Figure 366 shows the usual arrangement of condenser and auxiliaries in the older designs of piston-engine, low-level jet condensing plants where high vacua were of secondary consideration. Here each condenser receives its supply of cooling water from a main injection pipe and discharges into a main discharge pipe. The exhaust steam leading to the condenser is by-passed through a suitable atmospheric relief valve to a main free-exhaust header, so that the engine may operate non-condensing in case the vacuum breaks or the condenser is cut out. The wet-air pump is integral with the condenser chamber, and the entire installation is compact and simple. Occasionally conditions are such as to necessitate placing the condenser above the engine-room floor, as in Fig. 367, but such a location should be avoided if possible, as it usually requires a larger number of bends and joints in the exhaust pipe than the basement arrangement.

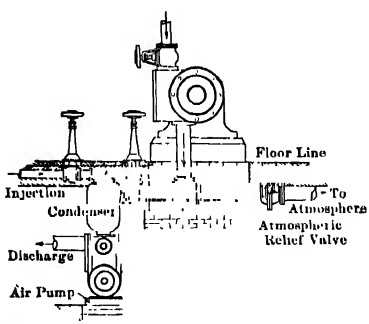


FIG. 366. Low-level Jet-condenser and Auxiliaries for M

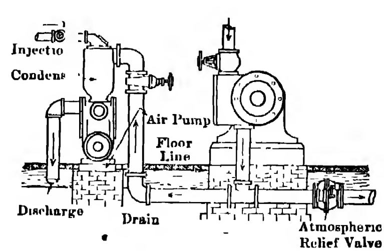


FIG. 367. Low-level Jet-condenser and Auxiliaries for Moderate Vacua.

Figure 368 shows the general arrangement of condenser and auxiliaries in the new power plant of the Johns-Manville Co. and illustrates a modern jet-condenser which has been installed in connection with a 2000-kw. turbine, and in which the air pump is of the hydraulic type. The con-

denser body is flexibly connected to the turbine by means of a corrugated-copper expansion joint. The back-outlet type of gate valve, placed between the expansion joint and condenser inlet, provides a means of cutting out the condenser and shunting the exhaust to the atmosphere should occasion arise. The tail pipe of the hydraulic air pump discharges into an open sump from which the air is liberated and the sealing water is recirculated.

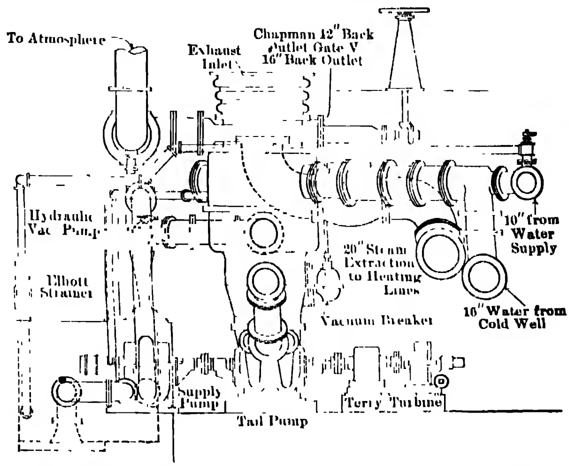


FIG. 368. Low-level Jet-condenser and Auxiliaries for High Vacua.

Figure 369 shows the general assembly of a C. H. Wheeler low-level jet-condenser as applied to a 2500-kw. turbine and illustrates the application of a steam ejector for air-vapor extraction. The centrifugal tail pump is the only moving element, and the simplicity of the entire condenser equipment is apparent from the drawing. The steam ejector has practically supplanted the other types of air pumps in the modern condensing plant. Figure 370 shows a typical layout of a 10,000-kw. jet-condenser with a combined turbine and motor driven tail pump and served with two jet air pumps, each of half capacity at 29-in. vacuum and therefore of sufficient capacity to carry full load at vacua of 28 in. This arrangement makes possible a 50 per cent saving in steam during the summer months with warm circulating water, when it is not possible to obtain much more than 28 in. on

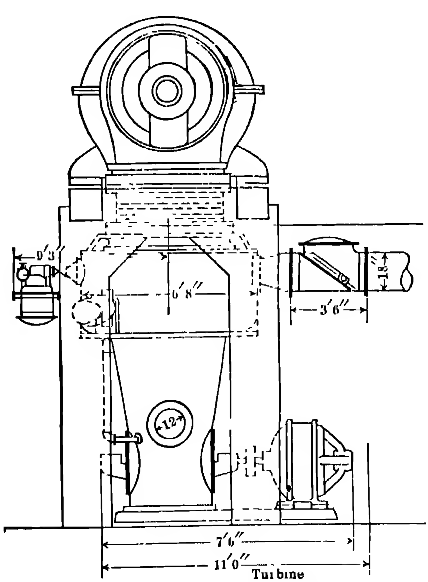


FIG. 369. Modern Arrangement of Low-level Jet-condenser and Auxiliaries for High Vacua.

the main unit and when, with warmer condensate temperature, no steam is required for heating the feedwater.

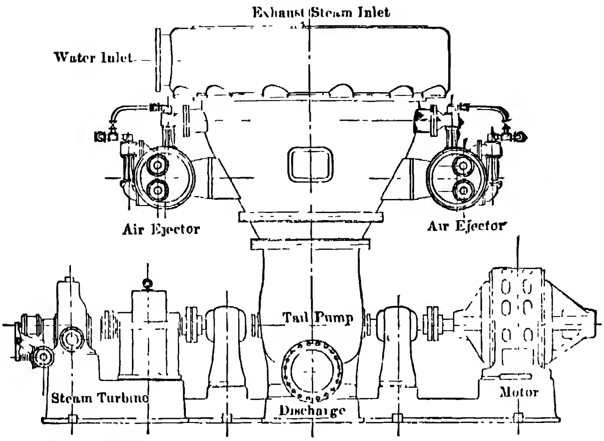


FIG. 370. Modern Low-level Jet-condenser Installation for Large Turbo-generator Units.

Figure 371 shows the general arrangement of the condensing equipment in the power house of the Bangor Electric Co., illustrating an application of a Koerting low-level multi-jet condenser to a 750-kw. turbo-generator above the turbine-room floor. This system operates without circulating or air pumps and maintains a vacuum of 28 in. with 70 deg. fahr. water under full-load conditions. Injection water is supplied from an elevated flume under

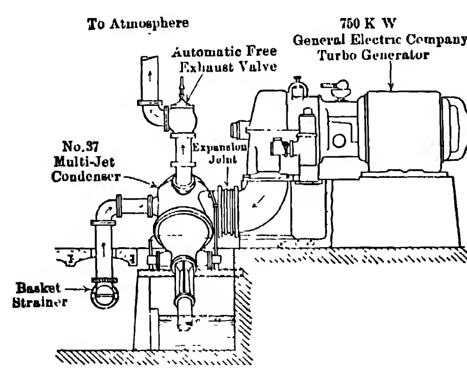


FIG. 371. Low-level Multi-jet Condenser Installation.

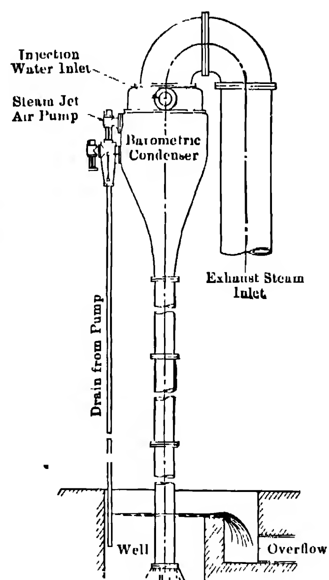


FIG. 372. Modern Installation of Barometric Condenser.

a natural head of 10 lb. and discharges under gravity from the hotwell to a creek.

Figure 372 shows a typical installation of a barometric condenser served with steam jet air pumps. The water from the inter-cooler (whether of jet or surface type) is ordinarily drained directly to the overflow well, the tail pipe being submerged, as indicated.

Figure 350 illustrates the usual layout of small surface condenser and auxiliaries where the heat balance justifies the use of direct-acting pumps. Figure 351 shows the more common arrangement in which the air pump

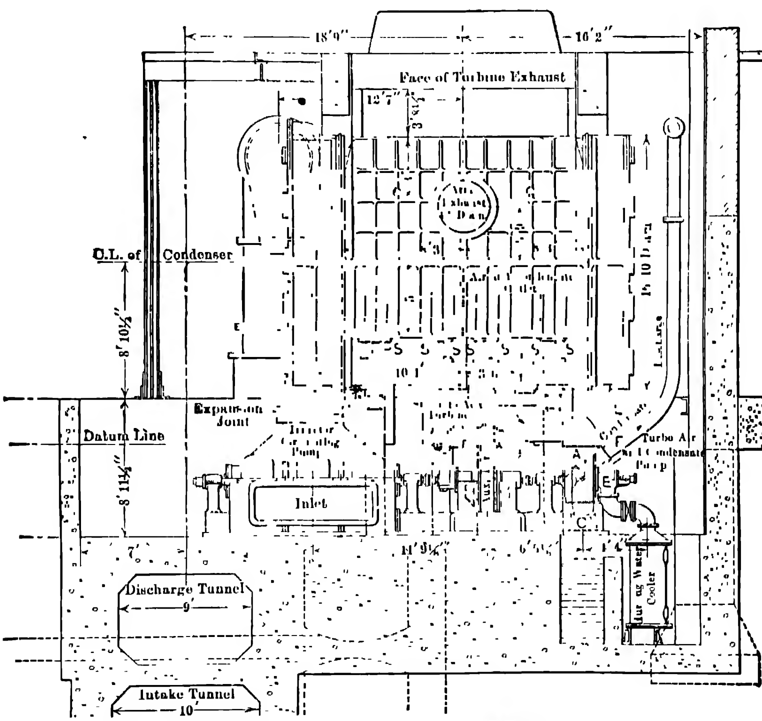


FIG. 373. Longitudinal Elevation of the 50,000 Sq. Ft. Condenser at "Northwest" Station.

is of the single-stage ejector type. In these small installations no provision is ordinarily made for expansion between condenser and engine or turbine, or between condenser and auxiliaries. With large surface condensers such provision is necessary. Figure 373 shows the arrangement of condenser and auxiliaries in the Northwest Station of the Commonwealth Edison, illustrating the method of compensating for expansion. The condenser is rigidly bolted to the turbine exhaust and is supported on a

number of heavy springs.[1] The circulating pump is flexibly connected to the condenser through a suitable expansion joint. The air pump is of the "hurling water" type.

Figures 374 and 375 show modern arrangements of surface condensers and auxiliaries in which the air removal is effected by steam jet pumps.

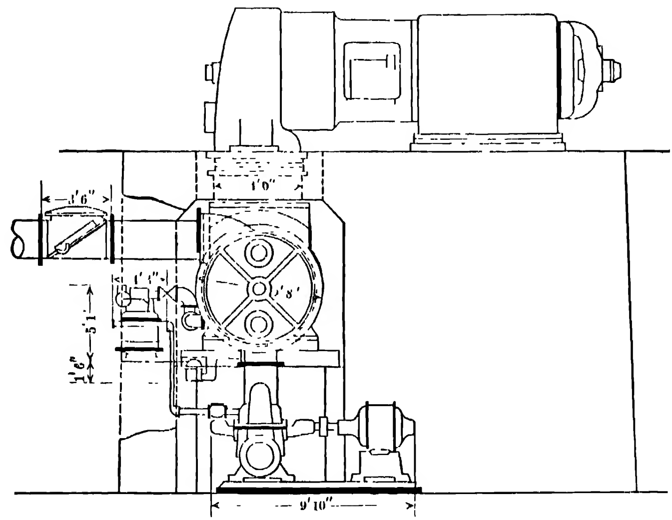


FIG. 374. Typical Arrangement of 2500-kw. C. H. Wheeler Turbo-generator Condenser and Auxiliaries.

In some designs the expansion joint is of rubber instead of corrugated copper as illustrated. With surface condensers, a siphon system of circulating should be installed where possible, with the highest point in the system not over 20-23 ft. above lowest water; and, if the condenser is installed directly above the inlet and outlet tunnels, the work of forcing the water through the system is limited to practically that of the friction through the tubes. With a siphon system, a rise in water level in both tunnels will not affect the pumping head and quantity of water delivered.

In some of the latest central station installations, of which the Crawford Ave. Plant is an example, the condensers are of the vertical type, two for each turbine unit, these condensers standing alongside the low-pressure cylinders.

Central condensing systems are not in evidence in the modern steam plant except in connection with steel mills or other industrial plants in which it is desired to operate condensing a number of comparatively

[1] Methods of Connecting Condensers to Turbines: Report of Prime Movers Committee, N.E.L.A., T5-21, 1921, p. 10.

small steam-using appliances or where the exhaust is intermittent. The particular advantage of this arrangement is the reduction in the number of auxiliary pumps and the prevention of loss of vacuum in case one or more steam units are not operating. This system is not used in connection with engine or turbo-generators.

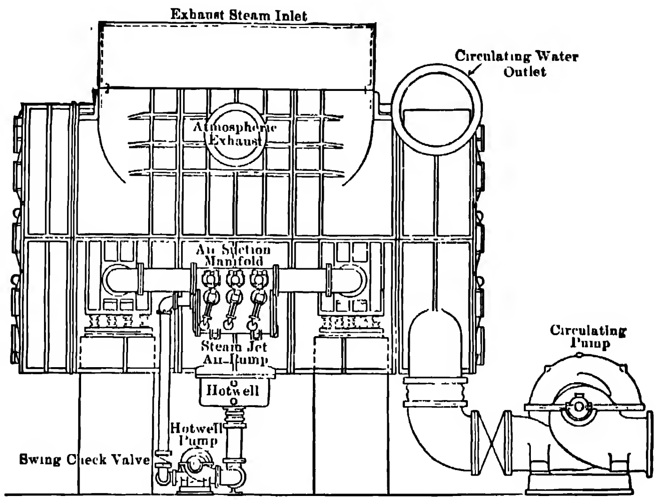


FIG. 375. 50,000 Sq. Ft. Condenser Layout with Three-element Wheeler Steam Jet Air Pump.

229. Choice of Condensers. — The proper selection of condenser and auxiliaries for a proposed installation depends upon the conditions under which the plant is to be operated. These conditions vary so widely in practice that only a few of the more important factors will be considered. The principal advantages and disadvantages of the three types of water-cooled condensers are as follows:

ADVANTAGES	DISADVANTAGES
Surface Condenser	
Re-use of condensate for boiler feed.	First cost high.
Re-use of condensate for ice production	Maintenance high.
Readily adapted to the weighing of condensate for tests.	Requires considerable building space to remove tubes.
Slightly better vacuum obtainable.	Acidulated water or water containing foreign matter in large quantities may preclude the use of surface condensers.
Advantage of low pumping head through siphon action.	More head room necessary to obtain sufficient head on hotwell pump.
Less chance of losing vacuum because a drop in vacuum does not affect water supply.	

Barometric Condenser

Condenser proper not costly, but piping to it is expensive.
No possibility of flooding turbine as in the case of a low jet condenser
Maintenance low.
The use of acidulated water possible.
Requires less circulating water than surface condenser.
Requires little building space
Equipment simple. No hotwell pump necessary and in some forms no vacuum pump is required.

Long exhaust pipe line to condenser which entails high initial cost and greater possibility of air leaks.
Loss of vacuum between turbine and condenser, which may amount to ½ inch or even more
As condenser cone generally extends above roof, it does not lend itself to economical station design when boiler room and turbine room are parallel and contiguous.
Waste of condensate.

Jet Condenser

Least expensive type of condenser.
Requires less building space.
Equipment simpler because hotwell pump is not necessary
Requires less circulating water than surface condenser.
Maintenance low.
The use of acidulated water possible.

Failure of removal pump would flood turbine. Protection is provided by a vacuum-breaking float valve.
Waste of condensate
High power for water pumping.
High power for air pump (about twice that for surface condensers).

Condenser auxiliaries are driven either by steam or electric motors or a combination of both. In the small power house which is not part of an inter-connected system and where the plant does not operate on a twenty-four-hour basis, steam-driven auxiliaries are, as a general rule, the best investment provided all the exhaust can be used for feedwater heating or other useful purposes. If there is more exhaust than can be utilized for feedwater heating or other purposes, part of the auxiliaries may be operated by duplex or combined steam and electric drives. In starting up, the steam unit is put into commission until current is available, when the governor automatically cuts in the electric drive. The circulating pump is usually of the steam-turbine-driven centrifugal type, though, in some of the very small plants, the reciprocating type is preferred. Steam jet air pumps are used in all but the very smallest plants and have practically supplanted the piston or hydraulic type. All condenser auxiliaries in these plants are ordinarily operated at full load irrespective of the load on the main unit, low first cost and simplicity of operation being of greater importance than heat economy.

In the large central station the type of drive is largely a matter of the station heat balance. (See paragraph 265.) While steam-driven auxiliaries are found in a number of recent designs, the present tendency is to rely more and more upon motors as a source of power for all auxiliaries. In order to insure against shut down of the main units through failure of power supply to the auxiliaries, the more important auxiliaries are supplied from a separate source of power, such as a house turbine, or are operated

by duplex drives. Duplication of auxiliaries is now practically standard practice with surface condensers for units of 15,000 kw. and over. Some of the more recent installations employ two constant-speed pumps on a single condenser having divided water-box construction. These pumps are provided with discharge valves as well as a by-pass valve, so that either pump may be used to supply water to the entire condenser, or each pump may supply water to one half of the condenser independent of the other half. Further economies have been obtained by the use of two-speed motors on either one or both of the pumps. The most efficient method of regulating the circulating supply is by the use of variable-speed drives for the pumps. Several recent condenser installations are provided with this means of control. Hotwell pumps are invariably of the centrifugal type, and are of either the single or two-stage type depending upon the design furnished by the condenser manufacturer. Owing to their reliability, low steam consumption, low maintenance cost, and compactness of installation, air pumps of the ejector type have practically supplanted the piston or hydraulic type of air pump. For a description of the various types of circulating condensate and vacuum pumps found in practice, see paragraphs 280 to 289.

Steam Condensing Plants: P A. Bancel, Trans. A.S.M.E, Vol. 43, 1921, p. 1051.

Relative Efficiency of Various Types of Condensing Apparatus. Brewer and Stivers, Mech. Engrg., Oct., 1921, p. 672; Power, Nov. 15, 1921, p. 1092.

Turbo-generators and Condensers for Modern Power Plants. Ganshird and Carothers, Elec. Rev., Sept. 17, 1923

Selecting Condensing Equipment for Power Plants. R. June, Elec. Rev., Sept. 17, 1921, p. 431.

Application of Steam Condensers: Selection of Size: F. A. Burg, Elec. Jour., Jan., 1921, p. 17.

Air Pump for Condensing Equipment: Frank R. Wheeler, Mech Engrg., Dec., 1919.

230. Water-cooling Systems. — When an ample supply of cooling water is unobtainable, for natural or economic reasons, the circulating water may be used over and over again by employing suitable cooling devices. The four systems most common in practice are:

1. The simple cooling pond or tank,
2. The spray fountain,
3. The cooling tower,
4. Coolers — surface type.

231. Simple Cooling Pond. — The simple pond is one of the oldest means of cooling and storing water for industrial and power plant purposes. The cooling action is independent of the depth of water and varies directly as the surface, the amount of heat dissipated for each sq. ft. of

exposed surface depending upon the temperature of the water, the temperature and relative humidity of the air, and the velocity of the air currents or wind. The maximum theoretical point of cooling is that of the temperature of the wet-bulb thermometer. The cooling action is effected by conduction and evaporation. Where the relative humidity is 100 per cent, all cooling is practically by convection and the amount of cooling is limited entirely by the amount of air that passes over the water to be cooled. Air is seldom quiescent, there being at all times some movement. Even if there is no wind blowing, the action of the heated water on the air would tend to create upward air currents and thus facilitate the action of cooling. Were this not so, there would be absolutely no cooling possible unless there were wind movement. When the relative humidity is less than 100 per cent — and this is invariably the case even during a heavy rain storm — part of the cooling is by evaporation. The amount of heat dissipated per sq. ft. of pond surface in perfectly calm air has been the subject of considerable experimental investigation, but the results have been decidedly discordant. Even if rules were available for calculating the amount of heat dissipated under these conditions, they would be of little service in determining the extent of surface necessary for practical installations because of the variable influence of air currents. For this reason engineers find it convenient to use rules of thumb which experience has taught will give satisfactory results. A common rule is to allow 6 to 8 sq. ft. of pond surface per lb. of water to be recirculated. Another, and perhaps more general rule, is to allow a heat transmission of 3.5 B.t.u. per hr. per sq. ft. of pond surface per degree difference in temperature between that of the air and the condenser discharge water. Since the heat is dissipated chiefly by evaporation, the weight of water evaporated is a fair index of the amount dissipated and approximates 1000 B.t.u. per lb. In the new plant recently installed by the Western Light and Power Co., Boulder, Colo., 3 sq. ft. of pond surface is allowed per 1000 B.t.u. per hr. to be removed from the circulating water.

Box gives the following formula for the rate of evaporation in perfectly calm air:

$$E = (243 + 3.7t)\ (V - v). \qquad (230)$$

in which

E = evaporation in grains per sq. ft. per hr.,
t = temperature of the water, deg. fahr.,
V = maximum vapor tension in in. of mercury at temperature t,
v = actual vapor tension.

A few scattering tests show that the evaporation as calculated from equation (230) should be increased 25 per cent for each mile per hr. of wind velocity.

Example 59. — How many lb. of water will be evaporated per sq. ft. per hr. from a pond, with the temperature of the water and air 80 deg. fahr.; air perfectly calm; barometric pressure 29.5 in. and relative humidity 70 per cent?

Solution. — The maximum vapor tension at temperature of 80 degrees is 1.03 in. of mercury. The actual vapor tension will be

$$1.03 \times 0.70 \ (= \text{relative humidity}) = 0.721.$$

Substitute these values in equation (230),

$$E = (243 + 3.7 \times 80)\ (1.03 - 0.721)$$
$$= 167 \text{ grains per sq. ft. per hr.} = 0.024 \text{ lb. per sq. ft. per hr.}$$

It will be noted that any one of the preceding rules gives an enormous pond surface for even a small cooling effect. For this reason the old-fashioned cooling ponds are seldom found in the modern plant except where ground space is inexpensive and the cost of excavation is low.

232. Spray Fountain. — To facilitate evaporation with a view toward reducing the size of the pond, the hot circulating water is generally distributed through pipes and discharged through nozzles, falling to the surface of the pond in a spray. The water issuing from the nozzles creates a draft which, aided by the natural breeze, effects the necessary evaporation. The loss of water due to evaporation seldom exceeds 4 per cent of the weight of water circulated. The pressure required at the nozzles is approximately 6 lb. per sq. in. and in many cases the condenser pump is able to furnish the necessary pressure. Under ordinary conditions the power necessary to operate the sprays will average less than 1 1/2 per cent of the power generated by the prime mover. Should the temperature of the condenser discharge-water exceed the limit of reduction by single spraying, the desired reduction in temperature may be effected by double spraying. In this arrangement, the condenser discharge is mixed in the hotwell with an equal amount of cooler water flowing through an equalizing valve from the spray pond. The resulting mixture is pumped to the nozzles and resprayed. Some idea of the performance of a spray cooling system may be gained from the data in Tables 71 and 72.

Natural ponds without sprays require about 50 times more area than spray cooling systems. A rough rule is to allow 130 B.t.u. per sq. ft. per hr. per degree difference in temperature for the latter.

TABLE 71

SINGLE-SPRAY SYSTEM — 6000-KW. STEAM TURBINE PLANT

Month	Relative Humidity Per Cent		Temperatures, Deg. Fahr.			
			8 A.M.	12 A.M.	4 P.M.	Remarks
Jan.	62	Discharge water..	68	73	73	Clear
		After spraying	48	53	53	
		Surrounding air .	8	14	20	
Mar...	50	Discharge water.. .	79	86	90	Clear
		After spraying.. .	58	66	70	
		Surrounding air .	30	50	43	
May	72	Discharge water..	89	94	97	Clear
		After spraying ..	70	75	78	
		Surrounding air.. ..	65	72	70	
July ..	70	Discharge water..	108	118	118	Clear
		After spraying .	90	93	93	
		Surrounding air .	90	98	102	
Aug. ...	81	Discharge water..	112	114	116	Cloudy
		After spraying	88	89	90	
		Surrounding air	72	74	79	
Nov. .	70	Discharge water.	89	90	88	Cloudy
		After spraying	62	64	63	
		Surrounding air	27	33	34	

TABLE 72

DOUBLE-SPRAY SYSTEM

	First Spraying	Second Spraying
Temperature air, deg. fahr.	87.0	88.0
Relative humidity, per cent	48.5	46.0
Temperature, hot water, deg. fahr .	122.5	88.7
Temperature, cooled water, deg. fahr. .	88.3	78.8
Total degrees cooled, fahr..		44.1

233. Cooling Towers. — While spray ponds require only a fraction of the area of the single cooling pond for the same refrigerating effect, the space is still considerable, and during periods of high winds a large quantity of water spray may be scattered over the surroundings and wasted. Aside from the monetary loss thus occasioned, there is the nuisance arising from the heat vapors or spray being deposited on neighboring buildings, walks, and streets. In order to reduce the space requirements and overcome these nuisances, the cooling tower has been developed. A cooling tower

consists of a wooden or sheet-iron housing, open at the top and bottom and so arranged that the hot water may be elevated to the top and distributed in such a manner that it falls in thin sheets or sprays into a reservoir at the bottom, air at the same time being drawn in at the bottom by natural draft or forced in by a fan. The water gives up its heat to the ascending current of air by evaporation, convection, and radiation, the last, however, being a relatively small factor. Of these, evaporation absorbs from 75 to 85 per cent of the heat, convection or direct transfer of heat to the air comes next, while radiation, partly in the tower and partly through the piping, accounts for the balance. If the air supply is dependent entirely upon the chimney action of the device, the system is known as a **natural-draft** or **flue cooling tower**; if the air is forced into the device by fans, the system is called a **forced-draft cooling tower.** Water-cooling towers may be classified as (1) forced-draft, (2) natural-draft — open type or atmospheric, (3) natural-draft — closed or flue type, and (4) combined forced and natural draft.

Forced-draft towers are completely enclosed, except at the top and at the base where provision is made for the fan openings. In the atmospheric type of natural-draft tower, the sides are louvered and the necessary air is supplied through the open base and through the louvered sides by natural air currents. The flue type of natural-draft tower receives its air supply through the chimney action of the flue. The combined forced- and natural-draft tower may be used with natural draft only for light loads and forced draft for heavy loads.

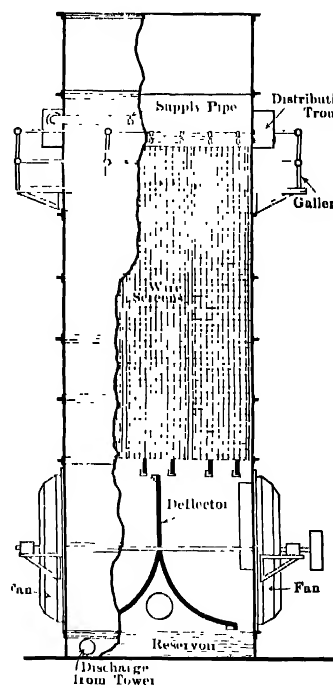


FIG. 376. Barnard-Wheeler All Steel Cooling Tower.

The different designs vary principally in the character of the filling and the method of water distribution. Figure 376 illustrates the Barnard-Wheeler cooling tower in which the falling water is broken up by vertically suspended galvanized iron wire-cloth mats, causing it to trickle in thin sheets to the bottom. In the Wheeler standard design, the tower is equipped with a large number of V-shaped horizontal wooden troughs arranged diagonally so that the spill from each trough is directed to the one immediately below. The ascending currents of air, in flowing through

the unrestricted zig-zag air lanes, mingle intimately with the slowly descending shower of water. The fillings of the Alberger standard cooling tower consist of boards of swamp cypress geometrically arranged in a honeycomb fashion, permitting the water to trickle down the sides of the boards and the air to pass upward through the lanes. In the C. H. Wheeler natural-draft cooling tower, Fig. 377, the filling is constructed of a series

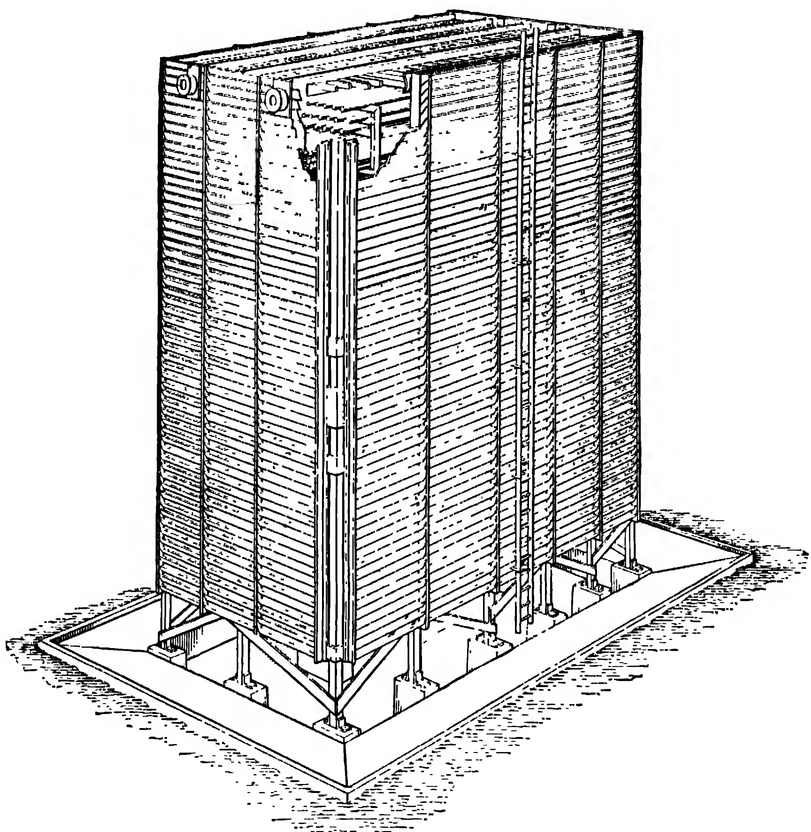


FIG. 377. C. H. Wheeler Atmospheric Cooling Tower.

of cypress strips laid in alternate rows and staggered so that no water can drop more than a few inches without being broken. The louvers are designed so that the air may pass freely into the structure and yet arrest the spray in a strong wind.

Owing to their compactness, cooling towers may be located on the roof of a power house or adjacent building, on the engine room floor or in the yard, the latter being the most adaptable. The minimum theoretical

temperature of the circulating water is that of the existing wet bulb, but in practice it is usually more economical to keep down the temperature range and circulate more water, the exact amount, of course, depending upon the nature of the equipment, load factor, and efficiency of pumping. The rate and degree of cooling is a function of the amount of water surface exposed to the air per unit of time and not a function of the amount of board or filling surface exposed to the water. As a general rule, spray ponds are installed where real estate is cheap and available, and cooling towers where space is expensive and unavailable. Cooling towers circulate less water than spray ponds because of their higher cooling efficiency, but the pumping head is higher, so that as far as power requirements are concerned there is no great difference between the two systems. See paragraph 109 for cooling tower calculations.

The Design of Cooling Towers Trans. A.S.M.E., Vol 44, 1922, p. 669.
Factors in Cooling Tower Design: Power, Feb 27, 1923, p 345
Water-Cooling System Efficiency: Mech. Engrg., Nov., Part 2, 1924, p. 799.

234. Coolers — Surface Type. — While circulating water for condenser purposes is never cooled in practice by coolers of the surface type, it is frequently necessary to employ such devices in the cooling of jacket water from internal combustion engines, cooling lubricating and transformer oils, cooling air for turbine generators operating on the "closed system," and for transferring the heat from one liquid to another. Coolers of the surface type are invariably used where the medium to be cooled is used over and over again without contact with the cooling medium. These devices are built in a great variety of designs ranging from the simple double-tube type in which the medium to be heated passes through the inner tube and the cooling medium passes through the annular space between both tubes, to the more complicated multi-tube or multi-compartment type in which the cooling surface is greatly extended so as to permit a high rate of heat exchange. For exchanging heat between liquids, the double-tube or multi-tube type is commonly used, but where one medium is a non-condensable gas and the other a liquid or gas, the extended-surface type is the more general because of the low rate of heat transmission.

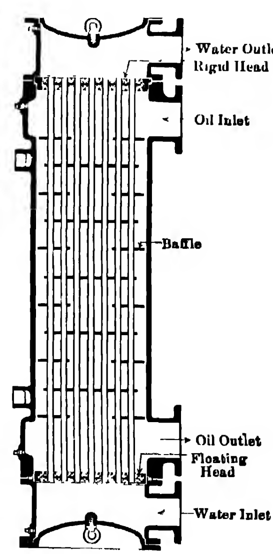


FIG. 378. Schutte-Koerting Oil Cooler.

Figure 378 shows a section through a Schutte and Koerting oil cooler and illustrates the principles of a popular make of cooler. The hot oil flows into the shell at the top, passes around the annular baffles, and

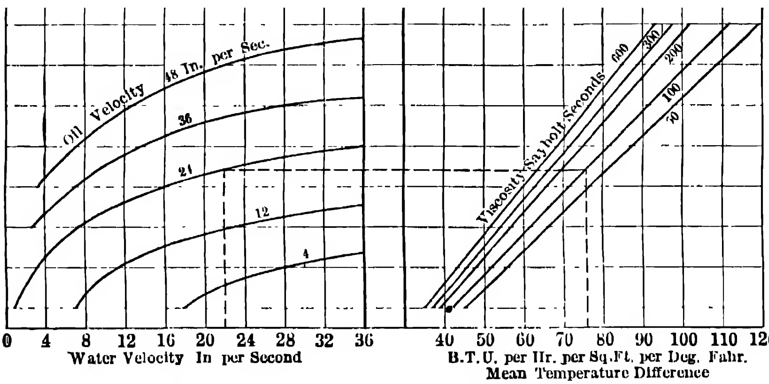


FIG. 378a. Heat Transfer — Schutte-Koerting Oil Cooler.

leaves the shell at the bottom. The water flows through the tubes in a counter-current direction. The heat-transfer rate for this type of heater is shown by the curves in Fig. 378*a*.

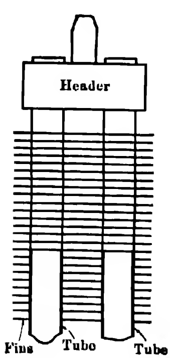


FIG. 379. "U-fin" Air-Cooler Element with Return Bend.

Figure 379 shows the principles of the Griscom - Russell Co.'s "U-fin Cooler" for cooling the ventilating air for turbo-generators. It differs from the ordinary type of smooth-tube cooler in that the external surface of the tubes is greatly extended by thin brass sheets which are in metallic contact with the tubes and which form a series of narrow channels for the passage of the air. The tubes are of Admiralty brass, 5/8-in. O.D., No. 18 B.W.G. gage and are spaced 1 9/16 in. between centers in each

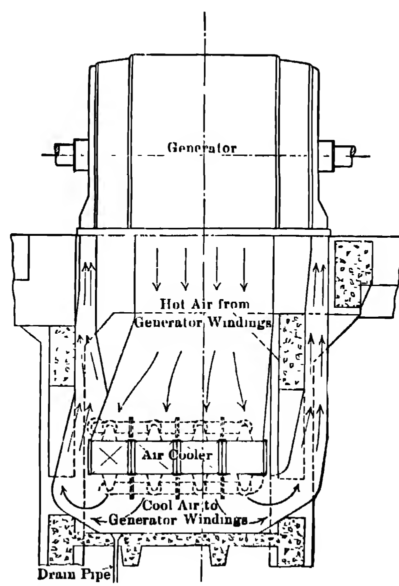


FIG. 379a. Typical Installation of "U-fin" Air Cooler to Turbo-generators.

element. The air fins are of brass, 30 B. & S. gage, and are secured to the tubing by tinning. The "U-fin Preheater" is built on the same principles as the cooler.

The coefficient of heat transfer from these various heat exchanging devices varies within such wide limits, depending upon the design of apparatus, physical properties of the heating and cooling medium, velocity of flow, initial temperatures, and the like, that the "average" values are only of academic value. Specific values may be had from the manufacturer. Valuable data in this connection may be found in the 1921–23 Prime Movers Committee Reports, N.E.L.A.

Ventilation of Turbo-alternators with Cool Purified Air: Power, Nov. 15, 1918, p. 921.
Heat Transmission in Coolers, Heaters and Condensers: Jour. Soc. Chem. Ind., Nov. 23, 1923, p. 443.

234a. Water Screens. — Cooling water, unless free from foreign matter, is apt to clog the orifices of jet condensers and the tubes of surface condensers, resulting in reduced efficiency of operation, increased load on boilers and stokers, and increased cost of fuel supply. Even when the water is comparatively free from foreign matter, it is customary to use some sort of screen to prevent fish or any chance obstruction from entering the circulating system. Stationary screens are simple and efficient, but require frequent cleaning. In large central stations or in small stations where the water is particularly bad, the traveling screen is accepted practice. The traveling screen consists essentially of two chains, passing around sprocket wheels at the head and foot, to which are attached wire cloth screens in the shape of steel trays or baskets. The trays are placed close together so as to form a continuous screen which travels with the chains. The foreign matter adheres to the surface of the screen and is dumped on the descending run into troughs, as shown in Fig. 380. Material which clings to the screen is washed off by water sprays under considerable pressure. The wire cloth is constructed either of copper, brass, steel, or galvanized wire according to the particular conditions of the circulating water. In the larger stations several screens are used so as to

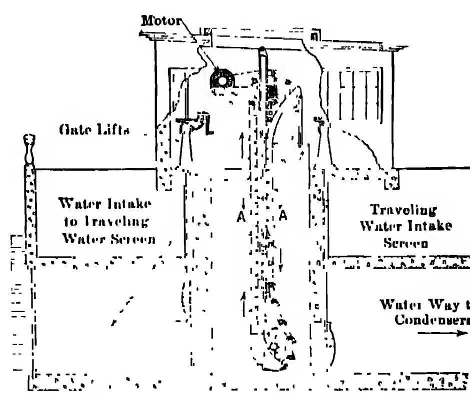


Fig. 380. Typical Traveling Water Intake Screen.

guard against interrupted service. The screens move very slowly, about 18 ft. per minute, so that very little power is required to drive them.

PROBLEMS

1. Reading of vacuum gage 26.5, temperature of room 80 deg fahr., barometer 29 5, temperature of mercury in the barometer 40 deg. fahr. Determine the vacuum referred to a 30-in. barometer.

2. If the absolute pressure in a condenser is 5 in. of mercury and the temperature of the air-vapor mixture in the chamber in 90 deg. fahr., required the percentage of air (by weight) in the mixture.

3. If the temperature within a condenser is 100 deg. fahr. and there is entrained 0.1 lb. of air per lb of steam, required the maximum degree of vacuum obtainable.

4. Required the volume of aqueous vapor to be withdrawn in order to cool 10,000 lb. of water from 120 to 80 deg fahr.

5. A 30,000-kw. turbine uses 12 lb. steam per kw-hr., initial pressure 290 lb. abs., superheat 250 deg. fahr., vacuum 28.5 in referred to a 30-in. barometer; initial temperature of the cooling water 70 deg. fahr., water velocity through tubes 8 ft. per sec. Required:

a. Weight of cooling water

b. Sq ft. condenser tube surface.

c. Number of 18 B.W G. 1-in. tubes in each pass of the condenser.

d. Length of water travel.

6. A 200-kw. turbine uses 20 lb steam per kw-hr., initial pressure 150 lb. abs., superheat 100 deg. fahr., vacuum 27 in referred to a 30-in. barometer If an evaporative surface condenser of the forced-draft type is used to create the vacuum, required the amount of atmospheric air and water spray which must be forced through the condenser. The temperature of the atmospheric air is 80 deg fahr., wet bulb thermometer 65 deg. fahr., air issuing from the condenser is completely saturated and its temperature is 15 degrees below that of the vapor in the condenser, fan pressure 4 in. of water.

7. How much "makeup" water is necessary for the cooling-tower system of a steam engine plant operating under the following conditions: Engines 1000 hp., water rate 20 lb. per i.hp-hr. initial pressure 120 lb. abs., vacuum 26 in, barometer 30 in.; temperature of injection water, discharge water and atmospheric air, 90, 110 and 70 deg. fahr., respectively; relative humidity of air entering and leaving tower 65 and 95 per cent respectively.

CHAPTER XIII

FEEDWATER TREATMENT, HEATERS, EVAPORATORS

235. General. — An ample supply of boiler feedwater of good quality is a necessity for economic and efficient operation of a steam plant. The larger the boiler units and the higher the rate of driving, the greater is the need for pure water. Among the numerous ill effects arising from the use of unsuitable feedwater may be mentioned (1) tube failures, (2) crystallization or embrittlement and corrosion of boiler steel, (3) loss of heat due to the deposit of scale, dirt, or oil on the heating surfaces, (4) length of time apparatus must be out of service for cleaning, inspection and repairs, (5) investment in spare equipment, (6) loss of heat due to blowing down boilers, heaters, etc., (7) increased steam consumption of prime movers due to accumulation of scale or dirt in valves, nozzles, and buckets, and (8) foaming and priming.

All natural waters contain more or less foreign matter either in suspension or solution; therefore, perfectly pure water can only be obtained by artificial treatment. Fortunately, pure water, while highly desirable, is not an economic necessity in all plants since the cost of purification may more than offset the gain due to elimination of all the ill effects previously outlined. This is particularly true in small or moderate-sized plants where the natural or raw water supply is of fairly good quality and where the boilers are not forced to any great extent and the service is not continuous. In plants of this class where the supply is poor, the water is usually treated for one or more seriously objectionable impurities, but no attempt is made to obtain the chemically pure product. In the large modern central station, with its tremendous output, extreme peak loads, and need for continuous operation, the quality of the feedwater is in many respects more important to the life and operation of the apparatus than is that of the fuel, and the expense of installing elaborate systems for purifying the water is usually warranted.

The impurities in water are determined by chemical analysis, and while such analyses are more or less standardized the formation of a correct conclusion is in many cases a difficult matter and is ordinarily beyond the powers of the layman. The impurities are usually determined in milligrams per 1000 liters of water, but are frequently reported as " parts by

weight per million parts of water by weight," "grains per standard U. S. gallon" or "pounds per 1000 lb. of water." Impurities which are electrically neutral and which do not enter into any combination in the water are weighed and reported as found, but salts in solution are determined **as ions** (calcium, sodium, chlorides, sulphates, etc.), and not as permanent salts (calcium chloride, sodium sulphate, etc.). Knowing the amount and character of the ions, the chemist is in a position to give the possible combinations of these ions in the form of salts. Since there is no way of proving from the analysis alone that any particular combination of the ions is formed to produce certain salts, rather than any other equally possible combination, it is customary to designate such combinations as **hypothetical combinations.** Engineers are accustomed to express the analyses in hypothetical combinations, since this method of reporting represents approximately the order in which precipitation takes place upon evaporation and enables them to visualize more readily the nature and amount of chemical treatment necessary. The more commonly found ions in feedwater and their hypothetical combinations are given in Tables 74 and 75.

The organic constituents of the foreign matter in raw water are of vegetable and animal origin and are taken up by the water in flowing over the ground or by direct contamination with sewage and industrial refuse. Feedwater containing organic matter may cause foaming, due to the fact that the suspended particles collect on the surface of the water in the boiler and impede the liberation of the steam bubbles arising to the surface. The inorganic impurities in suspension or in colloidal solution consist of clay, silica, iron, alumina, and the like. The more common soluble inorganic impurities are calcium, magnesium, potassium, and sodium in the form of carbonates, sulphates, chlorides, and nitrates.

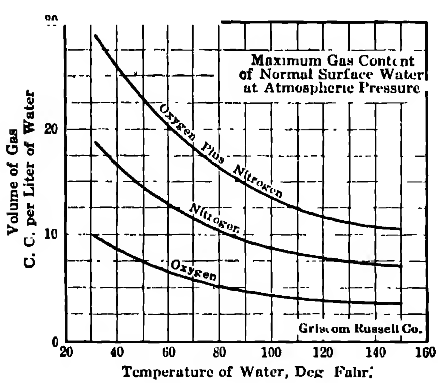


FIG. 381. Maximum Gas Content of Normal Surface Water.

Raw water also contains a certain quantity of gases in solution such as air,[1] CO_2, and occasionally hydrogen sulphide. Air, CO_2 and other gases may also be present in distilled water which has not been subjected to degasification. Gas-free distilled water absorbs oxygen and CO_2 at a

[1] Air in solution is usually designated as "dissolved oxygen" since the nitrogen content is inert and causes no trouble.

TABLE 73

WATER AND BOILER SCALE ANALYSES

Water Analysis. Grains Per U S. Gal.	Lake Michigan, Chicago	Well 115 Ft St Bernard, Ohio	San Francisco, Cal.	Schuylkill River, Philadelphia	Camaguey, Cuba	Park City, Utah	Toledo, Ohio	Kewanee, Ill.	Arkansas River, Florence, Colo	Surface Water, Auburn Park, Ill.
	1	2	3	4	5	6	7	8	9	10
Silica	0.438	0.677	0.759	0.338	2.873	1.354	0.759	0.373	0.630	0.508
Oxide of iron and aluminum	0.099	0.116	0.116	0.093	0.140	0.350	0.163	0.081	0.075	0.175
Carbonate of lime	3.731	2.271	4.207	0.068	10.270	1.476	5.519	1.721	2.158	2.382
Sulphate of lime	0.962	4.083	0.680	2.257	3.220	1.360	3.950	1.360	18.540	3.154
Carbonate of magnesia	2.092	4.424	2.866	0.884	4.900	0.318	2.592	2.212	4.848	2.875
Sodium and potassium sulphates	Trace	Trace	1.681	Trace	Trace	0.867	Trace	12.928	11.319	Trace
Sodium and potassium chlorides	0.670	0.990	2.970	0.990	5.708	1.980	2.740	26.070	2.028	1.650
Organic matter	0.066	0.584		0.700	33.000	2.569	1.052	0.584	0.701	0.584
Total mineral matter	8.058	12.614	13.665	4.672	32.288	7.82[illegible]	15.885	45.318	40.062	11.096
Chloride of magnesia					5.6					

SCALE ANALYSIS — PER CENT

	1	2	3	4	5	6	7	8	9	10
Character of sample	Hard, brittle	Medium hardness	Hard, brittle	Hard, impervious	Very hard	Very hard	Hard	Soft, brittle	Hard, crystalline	Medium hardness
Silica	20.60	8.44	11.18	12.30	24.42	19.00	4.96	2.52	6.20	5.7
Oxide of iron and aluminum	10.30	1.30	10.44	6.18	1.02	6.26	11.80	4.92	2.36	2.04
Carbonate of lime	33.86	37.22	40.96	21.26	29.10	29.02	3.74	18.18	18.78	29.86
Sulphate of lime	None	33.82	Trace	34.62	0.96	5.48	55.38	54.76	59.84	39.64
Carbonate of magnesia	6.04	Trace	22.60	Trace	Trace	Trace	8.19	Trace	0.84	Trace
Magnesia (MgO)	15.48	12.01		8.20	25.94	1.45	6.86	9.08	4.75	13.8
Moisture and organic matter	12.89	6.22	13.58	11.70	16.66	13.69	8.69	7.40	5.73	7.64
Oil	Trace			0.27				2.92		
Loss and undetermined	0.83	0.99	1.24	0.23	1.90	1.55	0.38	0.22	1.50	1.32
Lime (CaO)				5.24		23.55				

1. This water will cause the deposit of a moderate amount of scale which will be hard and persistent
2. This water will cause a large amount of scale to deposit.
3. This water will cause a moderate amount of scale with a decided tendency to galvanic action on account of the large proportion of sodium and potassium salts present
4. This water will cause the formation of a moderate amount of very hard scale.
5. This water will cause the deposition of a moderate amount of hard scale. The sodium and potassium salts together with the chloride of magnesia will induce galvanic action with consequent corrosion, pitting, etc.
6. This water will cause the formation of some scale. There is also a decided tendency to corrosive action
7. Will cause a hard and impervious scale to form
8. Will cause formation of some incrustation of medium hardness. It will also cause considerable trouble due to galvanic action, foaming and priming
9. This is not a desirable feedwater. It will cause the formation of considerable scale and will cause corrosion, pitting, and possibly foaming
10. Will cause the formation of a moderate amount of very hard scale.

very rapid rate and to a much greater capacity than water containing the usual mineral impurities in solution, so that the condensate from a high-vacuum condenser system, while practically free from gases, will absorb them at once on exposure. These gases, particularly oxygen, if permitted to enter the boiler or economizer with the feedwater, may under certain conditions cause excessive corrosive action on the steel heating surfaces.

TABLE 74

PROPERTIES OF IONS COMMONLY ENCOUNTERED IN FEEDWATER ANALYSIS AND PURIFICATION

Basic or Positive Ions			Acidic or Negative Ions		
Name of Ion	Symbol	Equivalent Weight	Name of Ion	Symbol	Equivalent Weight
Aluminium	$\overset{+++}{Al}$	9.03	Bicarbonate	$\overset{-}{HCO_3}$	61.01
Ammonium	$\overset{+}{NH_4}$	9.02	Carbonate	$\overset{--}{CO_3}$	30.00
Calcium	$\overset{++}{Ca}$	20.04	Chloride	$\overset{-}{Cl}$	35.46
Hydrogen	$\overset{+}{H}$	1.01	Hydroxide	$\overset{-}{OH}$	17.01
Ferrous	$\overset{+}{Fe}$	55.84	Nitrate	$\overset{-}{NO_3}$	62.01
Magnesium	$\overset{++}{Mg}$	12.16	Phosphate	$\overset{---}{PO_4}$	31.68
Potassium	$\overset{+}{K}$	39.10	Sulphate	$\overset{--}{SO_4}$	48.03
Sodium	$\overset{+}{Na}$	23.00			

In combining the positive and negative ions to form salts, there must be as many individual combining portions of negative ions as of positive ions. Thus:

$$\overset{++}{Ca} + (\overset{-}{Cl} + \overset{-}{Cl}) = CaCl_2; \quad (\overset{+}{Na} + \overset{+}{Na}) + \overset{--}{CO_3} = Na_2CO_3; \quad (\overset{+}{K} + \overset{+}{K} + \overset{+}{K}) + \overset{---}{PO_4} = K_3PO_4.$$

The product of the weight of any ion in milligrams per 1000 liters of water by the reciprocal of the equivalent weight gives the gram-equivalent. If the analysis is correctly made the sum of the gram-equivalents of the positive ions will be equal to the sum of the gram-equivalents of the negative ions.

Atomic weight = equivalent weight × valence (valence = number of + or − signs above the symbol). Thus: Atomic weight of aluminum = 9.03 × 3 = 27.09; that of calcium = 20.04 × 2 = 40.08; that of chloride = 35.46 × 1 = 35.46.

When raw water is fed into a boiler, practically all of the solids remain in the boiler and are constantly increased in amount by the evaporation taking place. Some of the accumulated impurities deposit on the heating

surface as scale, some are present as suspended matter, and others remain in solution. The dissolved gases are set free and the greater part is discharged with the steam. The remaining portion under certain conditions forms a number of combinations with the iron walls which result in pitting and general corrosion.

TABLE 75

COMBINATIONS OF IONS COMMONLY ENCOUNTERED IN BOILER FEEDWATER ANALYSIS AND PURIFICATION

Name of Combination	Symbol	Molecular Weight	Name of Combination	Symbol	Molecular Weight
Alumina	Al_2O_3	102.2	Magnesium bicarbonate	$Mg(HCO_3)_2$	146.3
			carbonate	$MgCO_3$	84.3
Aluminum hydroxide	$Al(OH)_3$	78.1	chloride	$MgCl_2$	95.2
sulphate	$Al_2(SO_4)_3$	342.2	hydroxide	$Mg(OH)_2$	58.3
			sulphate	$MgSO_4$	120.3
Ammonium bicarbonate	NH_4HCO_3	79.0			
carbonate	$(NH_4)_2CO_3$	96.0	Potassium carbonate	K_2CO_3	138.2
chloride	NH_4Cl	53.5	chloride	KCl	74.5
sulphate	$(NH_4)_2SO_4$	114.0	hydroxide	KOH	56.1
			nitrate	KNO_3	101.1
Barium carbonate	$BaCO_3$	197.4	sulphate	K_2SO_4	174.3
chloride	$BaCl_2$	208.3			
hydroxide	$Ba(OH)_2$	171.4	Silica	SiO_2	60.3
sulphate	$BaSO_4$	233.4			
			Sodium aluminate	$Na_2Al_2O_4$	144.1
Calcium bicarbonate	$Ca(HCO_3)$	162.1	bicarbonate	$NaHCO_3$	84.0
carbonate	$CaCO_3$	100.1	carbonate	Na_2CO_3	106.0
chloride	$CaCl_2$	75.5	chloride	$NaCl$	58.5
hydroxide	$Ca(OH)_2$	74.1	fluoride	NaF	42.0
oxide	CaO	56.1	hydroxide	$NaOH$	40.0
phosphate	$Ca_3(PO_4)_2$	310.3	nitrate	$NaNO_3$	85.0
sulphate	$CaSO_4$	136.1	phosphate	Na_3PO_4	164.0
			silicate	Na_4SiO_4	152.3
Ferrous carbonate	$FeCO_3$	115.8	sulphate	Na_2SO_4	142.0
sulphate	$FeSO_4$	151.8			
Ferric oxide	Fe_2O_3	159.7			

The most widely known evidence of the presence of scale-forming ingredients in feedwater is known as **hardness.** If the water contains only such ingredients as the bicarbonates of lime, magnesia, and iron, which may be precipitated as normal carbonates by boiling at 212 deg. fahr., it is said to have **temporary hardness. Permanent hardness** is due to the presence of sulphates, chlorides and nitrates of lime, magnesia, and iron which are not completely precipitated at a temperature of 212 deg. fahr. Hardness is conveniently determined by means of a standard soap solution as follows:

A 100-cc. (cubic centimeter) sample of water to be tested is put in a

250-cc. bottle and a standard soap solution (this may be obtained from chemical dealers) run in 0.2 cc. at a time, the bottle being shaken vigorously after each addition of the soap solution. Finally a lather is produced that will persist for at least five minutes, and then the volume of soap solution used in cc. gives the degrees "U. S." hardness. One degree "U. S." hardness is equivalent to 1 grain of calcium carbonate per U. S. gallon (1 part in 58,349).

The following factors may be used for specifying hardness of water in terms of calcium carbonate per U. S. gallon:

Magnesium carbonate	× 1.19	
Magnesium sulphate	× 0.833	
Calcium sulphate	× 0.735	= hardness as calcium carbonate, grains
Magnesium chloride	× 1.05	per U. S. gallon or U. S. degrees.
Calcium chloride	× 0.901	

It is impossible to judge the quality of feedwater merely by the grains of solids per gallon, since a large amount of soluble salt, such as sodium chloride, will not be as deleterious as a very small amount of calcium sulphate.

The scale of hardness usually accepted (grains of dissolved salts per U. S. gallon) is as follows: Soft water, 1 to 10; moderately hard 10 to 20; very hard water, above 25.

The following is a rough rating according to the number of grains of incrusting solids per United States gallon:

Less than

8 grains	very good.
12 to 15 grains	good.
15 to 20 grains	fair.
20 to 30 grains	bad.
Over 30 grains	very bad.

This applies to calcium carbonate, magnesium carbonate, and magnesium chloride. For water containing sulphates of calcium and magnesium, divide the first column by 4 for the same rating.

The limiting factor in deciding whether a water carrying a large amount of non-corrosive soluble salts may be used for boiler feed purposes is the amount of blowing down necessary to keep the degree of concentration within the limits found by experience.

The degree of concentration may be ascertained by a complete chemical analysis, but this is usually an expensive procedure and requires considerable time. The total solids in a given water of varying concentration generally bear a certain constant ratio to the sodium chloride content,

therefore, any method of determining the amount of sodium chloride in the water to be tested offers a satisfactory check on the total amount of solids present. The usual test for sodium chloride is to titrate a sample of the water in question with a normal silver nitrate solution, using potassium chromate as an indicator (see N.E.L.A. Report T3-22, 1922, p. 189). The **Esterline-Angus Concentration Meter** is finding favor with many engineers. This apparatus indicates or records the degree of concentration by measuring the variation in conductivity of the water. From experiments conducted under the supervision of the U. S. Bureau of Mines, it appears that if the proper relation of sulphate and carbonate, or sulphate and phosphate concentration is maintained at all times in the boiler water, there will be no growth of adherent scale on the heating surfaces. No feedwater analysis is necessary other than to test it for acidity. The concentration of carbonate, sulphate and phosphate radicals is readily determined by suitable titration of a sample of water drawn from the boiler. A knowledge of this concentration in conjunction with established curves is all that is necessary to properly condition the water so that hard adherent scale will not deposit on the heating surfaces. For a complete discussion of this important topic consult *Fundamentals in the Conditioning of Boiler Waters* by R. E. Hall: Proc. Engr. Soc. Wes. Pa., Vol. 41, Sept. 21, '25.

Boiler Waters: Their Chemical Composition, Use and Treatment. Univ. of Tex., Bul., 1752, Sept. 15, 1917

Treatment of Water for Steam Making. Chem. Age, Jan., 1922, p. 43.

A Review of Feedwater Treatment Power, Dec. 26, 1922, p. 1018.

Causes of Hardness in Water and How to Find Them Power Plant Engrg, Jan. 1, 1923, p. 36.

The ill effects from the use of impure feedwater may be briefly summarized as follows:

1. Scale, or incrustation.
2. Corrosion.
3. Metal embrittlement.
4. Foaming.

236. Scale. — Mud or suspended mineral matter, if introduced into the boiler with the feedwater, will eventually form a deposit on the heating surfaces. Iron, aluminum, and silicon in colloidal solution will also tend to produce scale, but by far the greater part of the objectionable scale deposit results from the salts of calcium and magnesium. The salts are in solution in the cold raw water and constitute "hardness." When subjected to the temperature and pressure in the boiler and to concentration by evaporation, certain portions are precipitated and form sludge or scale. The carbonates of calcium and magnesium alone usually produce scale of a chalk-like formation, which is more or less friable, but in the presence of other elements the formation may be hard and dense. Mag-

nesium sulphate alone does not form a hard scale, but when combined with calcium carbonate may produce a hard stony deposit. Calcium sulphate alone produces a hard tenacious scale. It is very difficult to tell from the composition of the water whether the scale produced will be adhering, non-adhering, hard, or soft. The type of boiler, pressure, degree of concentration, and rate of driving are all factors which influence the character and amount of deposit. Other salts, such as sodium carbonate, sodium chloride, and magnesium chloride, are always in solution and do not form scale except under excessive concentration.

Scale lowers the efficiency of heat transmission by insulating the surface of metal and reduces the capacity of the boiler. Overheating, with consequent blistering or bagging of tubes and shell, is frequently due to the heat-insulating effects of scale. Numerous tests on the heat transmission through boiler tubes coated with scale of varying thickness fail to show any relationship between thickness of scale and efficiency of transmission, but that even the thinnest coating of scale appreciably reduces boiler efficiency and capacity is too well known to dwell upon.

In plants using raw water for boiler feed, scale is removed periodically by "cutting out" the boiler and running rotary cleaners through the tubes. Chemical compounds fed into the boiler may reduce the number of cleanings, or in some cases dispense with mechanical cleaning entirely. Scale may also be prevented from entering the boiler by suitable treatment of the water outside the boiler.

Removing Boiler Scale with Co_2 Power, Mar. 14, 1922, p 422.

237. Corrosion. — Corrosion, both internal and external, is evidenced by small pits or depressions and by large cup-shaped hollows on the metal surface, and occasionally by a considerable destruction of a large portion of the surface. Carbonic acid gas, occluded oxygen, sodium and magnesium chlorides, and iron and aluminum sulphates are common causes of internal corrosion. Magnesium and calcium chlorides are very pernicious in that they produce free hydrochloric acid on hydrolysis. Galvanic action set up by any difference in the crystalline structure or chemical composition of the steel in different parts of the boiler will cause corrosion. Corrosion is also found in boilers using a high percentage of condensate or distilled water from which the dissolved gases have not been eliminated, or which has re-absorbed them on exposure to air, or which has become contaminated with raw water through condenser leakage. Whatever may be the theory involved in the corrosive action of dissolved gases in feedwater, it suffices to state that boilers or economizers fed continuously with pure condensate containing dissolved gases are subject to corrosion and that no such action is evidenced if the gases are removed and the

condensate is maintained slightly alkaline. Scaling and corrosion are sharply differentiated in that, with the exception of scales consisting mainly of silicates, one rarely accompanies the other. In fact, scale, other than silicate, is an excellent preventive of corrosion though decidedly objectionable in other respects. The test generally used for determining the oxygen content of feedwater is known as the **Wrinkler[1] method,** and the degree of acidity or alkalinity may be determined by a complete chemical analysis, by observing the color reaction when certain indicators are added, or by noting the voltage existing across two electrodes immersed in a small by-passed flow of the feedwater. (Consult "A Meter for Recording Alkalinity of Boiler Feedwater," by Robert E. Arthur and Earl A. Keeler, Power, May 16, 1922, p. 768.)

For an excellent discussion of the subject of corrosion and feedwater treatment in general, consult pages 164 194 Report of Prime Movers Committee, National Elec. Light Assoc. T3 22, 1922.

238. Embrittlement of Boiler Metal. — Embrittlement with consequent cracking of the metal in the seams is evidenced in boilers fed with artesian well water abnormally high in free sodium bicarbonate, and in plants where treatment with soda ash, caustic soda, or boiler compound has been carried to excess. It has been definitely established that concentrated caustic in the presence of steel will liberate free hydrogen gas and that the steel is susceptible of occluding the gas with a subsequent embrittling action. It is believed, therefore, that such action takes place in boiler joints where caustic soda, formed by the hydrolysis of sodium carbonate and sodium bicarbonate under temperature and pressure, is likely to concentrate. All waters containing free caustic or substances which produce excess causticity are not necessarily dangerous, but excessive concentration should be avoided by periodic blowing down of the boilers.

The Embrittling Action of Sodium Hydroxide on Soft Steel. S. W. Parr, Bul. No. 94, Univ. of Illinois. See also pp. 180-184, Report T3-22, 1922, N.E.L A.

239. Foaming and Priming. — Foaming is usually caused by certain types of organic matter, saponifiable oils in the presence of caustic soda or sodium carbonate, and suspended matter. It is largely a matter of viscosity of surface films, and surface blowing is a remedy where it can be applied. Foaming caused by organic matter in suspension may be minimized by filtration. Priming is due to increased surface tension, which tends to liberate the steam in slugs. The point at which priming occurs varies with different waters, different boilers, and the rate at which the boilers operate; but if the priming point is once determined and concen-

[1] Report T3-22, 1922, N.E.L.A., p. 168.

tration is maintained just below that point, little trouble will be experienced. Foaming and priming cause the impurities in the entrained boiler water to be carried over with the steam into the superheater pipe lines, traps, and prime movers, resulting in all of the troubles arising from the use of dirty apparatus.

Priming: Power Plant Engrg, May 1–15, 1922, pp. 456, 511; National Engr, Nov., 1920, p. 532; Power Plant Engrg., Apr. 1, 1925, p. 377.

240. Feedwater Treatment. — An ideal feedwater supply is one that will not deposit mud or scale, will cause neither priming nor foaming, and will not corrode boilers or appurtenances. No such water exists in the natural state, although many waters are sufficiently low in impurities to warrant their use, under certain conditions, in the raw state without purification. The deciding factor lies in whether the cost of maintenance and operation with raw water is greater or less than that incident to the use of the treated product. The quality of the feedwater plays such an important part in the economic operation of the steam plant that advice from a competent water-treating engineer is essential even in the smallest plants. In some plants raw water gives satisfactory results; in others partial treatment is necessary; while in some of our largest stations total elimination of all impurities is essential. There is no general panacea for treatment, and each installation and source of supply must be analyzed to meet the particular conditions involved.

All or part of the evil effects arising from the use of impure feedwater may be neutralized or eliminated by one or more of the following methods:

1. Filtration.
2. Preheating.
3. Chemical treatment.
4. Application of protective coatings.
5. Distillation.
6. Degasification.

Table 76, based on a similar chart by W. W. Christie, gives a general outline of the troubles arising from feedwater, their cause, and some of the means for preventing them.

241. Filtration. — Suspended matter, either in raw or treated feedwater, is cheaply and conveniently removed by passing it through a filter. There is a large variety of straining and filtering equipment on the market, but the down-flow type of filter, using sand or granulated quartz as a separating medium, appears to be the most common. Frequently a large part of the impurities in a water supply can be removed by filtration. Filters should be of ample size for service required; otherwise they soon become choked up or permit some of the filtering medium to pass into the water system. Mud and sand may under certain conditions be eliminated by

TABLE 76

BOILER TROUBLES ARISING FROM USE OF IMPURE FEEDWATER

Trouble	Cause	Remedy or Palliation
Incrustation	Sediment, mud, clay, etc.	Filtration Blowing off
	Readily soluble salts	Blowing off
	Bicarbonate of magnesia, lime, iron	Heating feed and precipitation Caustic soda Lime Zeolite
	Organic matter	See below
	Sulphate of lime	Sodium carbonate Zeolite Barium chloride
Corrosion	Organic matter	Precipitation with alum Precipitation with ferric chloride and filtration
	Grease	Slaked lime Carbonate of soda and filtration
	Chloride or sulphate of magnesium	Carbonate of soda
	Sugar Acid	Alkali
	Dissolved carbonic acid and oxygen	Slaked lime Caustic soda Heating Deactivator Deaerator
	Electrolytic action	Zinc plates
Priming	Sewage	Precipitation with alum or ferric chloride and filtration
	Alkalies	Heating feed and precipitation
	Carbonate of soda in large quantities	Barium chloride
Foaming	Organic matter Saponifiable oils in presence of caustic soda or sodium carbonate Suspended matter	Surface blowing Filtration
Embrittlement	Abnormal causticity	Magnesium sulphate Blowing off

simply permitting the water to stand for some time in settling tanks. It is unnecessary to filter water before chemical treatment except when excessively turbid, since the chemicals in the reaction tank act to some extent as coagulants and the suspended matter, whether originally contained in the water or produced by the chemicals, is eliminated by sedimentation or by filtration after treatment.

The **Hagan de-concentrator,** which consists of a special pump and filter, and prevents scale by continuously removing the insoluble impurities from the feedwater after it has been fed into the boiler, is finding increasing favor with engineers. By means of this apparatus water is withdrawn

from the boiler, preferably at the lowest point, forced through a suitable filter, and then returned. At intervals of twenty-four hours the filter is flushed back, and the impurities precipitated by the temperature in the boiler are washed out and discharged to waste.

Tests with Hagan De-concentrators: Prime Movers Committee, N.E.L.A., Report, 1922, Part B, p. 186.

Pressure Filters: Jour. Am. W. Wks. Assoc., Vol. 3, No. 2, 1916.

242. Preheating. — Practically all of the dissolved air and free carbonic acid, CO_2, in water may be liberated from the water by boiling it violently at 212 deg. fahr., under atmospheric pressure. If the liberated gases are removed by suitable means and the water is not exposed to further absorption of these gases before being fed into the boiler or economizer, there will be little danger from corrosion provided there are no other corrosive agents in the water. The bicarbonates of calcium and magnesium, which constitute the chief source of hardness in most boiler feedwaters, are broken up into carbonates and CO_2 when the water is heated to 212 deg. fahr. The reaction is as follows:

$$Ca\,(H\,CO_3)_2 = CaCO_3 + CO_2 + H_2O \tag{231}$$

$$Mg\,(H\,CO_3)_2 = MgCO_3 + CO_2 + H_2O \tag{231a}$$

The calcium carbonate is practically insoluble in the hot water and is precipitated as a solid, but the magnesium carbonate is only partly precipitated, since it is somewhat soluble. A large portion of the CO_2 is liberated and may be withdrawn with the other gases freed by boiling. While calcium carbonate is more soluble in hot water than in cold water, the difference is negligible and the greater part of the hardness that is due to its presence may be removed by boiling at atmospheric pressure.

Thus we see that nearly all of the dissolved gases and some of the scale-forming elements in water may be eliminated by merely boiling it violently at 212 deg. fahr., under atmospheric pressure and withdrawing the liberated gases. In the standard commercial type of exhaust steam heater, provision is not ordinarily made for a complete removal of the liberated gases, and the time the water is in the apparatus is not sufficient to allow all the precipitated matter to collect. There is no question, however, but that these devices have a decided purifying action.

Calcium sulphate, under the high pressure and temperature found in current practice, is practically insoluble, and unless the water is properly conditioned the precipitation will collect on the boiler heating surfaces as a hard, tenacious scale. (See Live Steam Purifiers, paragraph 259.)

See also paragraph 247.

243. Chemical Treatment. — The great majority of plants using treated water for boiler feed purposes depend upon chemical treatment for effecting the desired results. It should be stated at the outset that such treatment does not produce pure water, and as a matter of fact, frequently increases the total amount of impurities, but the objectionable impurities have been converted by this treatment into others which are less objectionable. Thus, when soda (Na_2CO_3) is fed into a boiler, the water of which has calcium sulphate ($CaSO_4$) in solution, the mineral content of the boiler has been increased by the amount of soda introduced, but the calcium sulphate, which produces a hard, tenacious scale, has been converted by the reaction with the soda into calcium carbonate, $Ca\,CO_3$, and sodium sulphate, Na_2SO_4. The calcium carbonate is practically insoluble in the hot boiler water and is precipitated as a sludge, so that it can be readily removed by blowing off. The sodium sulphate remains in solution, and produces no scale except under excessive concentration. If soda is added to the water before it enters the boiler the same chemical reaction takes place as within the boiler, but the precipitated calcium carbonate may be removed by sedimentation or filtration, and only the sodium sulphate will be introduced into the boiler. The water in this last case is said to be "softened," that is, the hardness due to the calcium sulphate has been eliminated. Whatever may be the process employed, the "purified" product usually contains an excess of alkaline salts and is far from being chemically pure.

If the nature and the amount of all the impurities in the raw water are known (this can only be determined by a complete mineral analysis), the chemist is in a position to specify the kind and the amount of reagent necessary to effect certain results. As previously stated, the impurities are determined by the chemist as ions, and the amount of reagents to be added is usually calculated from the ion content by use of proper factors. Engineers, on the other hand, prefer to have the ions grouped as hypothetical compounds. Whatever may be the method of procedure, the results will be the same provided the hypothetical compounds are properly grouped. Water analysis and purification is a highly specialized art and ordinarily beyond the province of the non-chemical engineer, but an idea of the method of procedure in calculating the weight of reagent necessary for a given chemical reaction may be gained from the following:

The chemical changes which take place when hydrated lime, $Ca(OH)_2$, and soda ash, Na_2CO_3, alone or in combination with each other, are added to water containing calcium sulphate, $CaSO_4$, magnesium sulphate, $MgSO_4$, calcium bicarbonate, $Ca(HCO_3)_2$, or magnesium bicarbonate, $Mg(HCO_3)_2$, may be expressed:

$$CaSO_4 + Na_2CO_3 = CaCO_3 + Na_2SO_4 \quad (232)$$
$$MgSO_4 + Ca(OH)_2 + Na_2CO_3 = Mg(OH)_2 + CaCO_3 + Na_2SO_4 \quad (233)$$
$$Ca(HCO_3)_2 + Ca(OH)_2 = 2CaCO_3 + 2H_2O. \quad (234)$$
$$Mg(HCO_3)_2 + 2Ca(OH)_2 = Mg(OH)_2 + 2CaCO_3 + 2H_2O. \quad (235)$$

From these reactions the amount of reagent to be added to raw water may be calculated by considering the combining weights as follows:

For soda ash and calcium sulphate

$$CaSO_4 : Na_2CO_3 = 1 : x,$$
$$40 + 32 + 4(16) : 2(23) + 12 + 3(16) = 1 : x,$$
$$x = 0.779,$$

in which

x = soda-ash factor or the ratio of the weight of soda ash required to the weight of calcium sulphate in the water.

By similar calculations the factors for salts which require soda ash are found to be as in Table 77.

The chemist usually calculates the weight of reagent directly from the ion content because the analysis is expressed in ions, but the results are practically the same as when calculated from hypothetical combinations. (Consult *Analytical Control of Water Softening*, Univ. of Ill. Bulletin, Vol. 8, No. 23, pp. 88–148. *Boiler Waters: Their Chemical Composition, Use and Treatment*, Univ. of Tex. Bulletin, No. 1752.)

While soda ash and lime are the most commonly used reagents for softening water because of their availability and cheapness, numerous other substances may effect the same result.

Chemical Treatment of Feedwater: Power, Dec. 19, 1922, p. 984.
A Review of Feedwater Treatment: Power, Dec. 26, 1922, p. 1018.
Relation of Water Purification to Boiler Operation: Nat. Engr., Nov., 1920, p. 532.

244. Boiler Compounds. — When the reagents are added to the feedwater or introduced directly into the boiler and the reaction takes place within the boiler itself, the process is commonly designated as treatment by **boiler compound.** A great variety of substances have been employed for this internal treatment. Among them may be mentioned soda ash, lime, barium hydroxide, sodium silicate, sodium aluminate, tannin, dextrine, trisodium phosphate, and the like. Many of the patented compounds are worthless and actually aggravate the trouble which they are supposed to remedy, but, taking all things into consideration, the use of a suitable compound is probably the least expensive form of feedwater treatment in moderate-sized plants where the water contains a small amount of scale-forming elements and where the rate of driving is not

high. The ingredients in the compound should be based on the feedwater analysis, and under no circumstances should an unknown substance be introduced into the boiler. The most satisfactory compounds are those which not only effect a precipitation of the scale-forming ions through chemical action but also render the precipitated matter non-adherent by mechanical action. Sodium aluminate, tannates in conjunction with lime and soda, and sodium silicate produce such results in waters suitable for this treatment. The "Navy Standard Boiler Compound" is a well-known example of this class of reagent and is composed of 76 per cent anhydrous sodium carbonate, 10 per cent trisodium phosphate, 1 per cent dextrine or starch, and sufficient cutch to yield at least 2 per cent of tannic acid, the balance being water. Sodium aluminate alone has given excellent results with water from the Great Lakes and the rivers of the Mississippi Valley.

TABLE 77

FACTORS FOR USE WITH HYPOTHETICAL COMBINATIONS

Salt	Factor		
	Soda Ash Na_2CO_3	Lump Lime CaO	Hydrated Lime $Ca(OH)_2$
Calcium chloride, $CaCl_2$	0.955		
Calcium sulphate, $CaSO_4$	0.779		
Calcium carbonate, $CaCO_3$		0.560	0.740
Calcium bicarbonate, $Ca(HCO_3)_2$		0.346	0.457
Magnesium chloride, $CaCl_2$	1.113	0.589	0.778
Magnesium sulphate, $MgSO_4$	0.881	0.466	0.616
Magnesium carbonate, $MgCO_3$		1.330	1.757
Magnesium bicarbonate, $Mg(HCO_3)_2$		0.767	1.014
Sodium carbonate, Na_2CO_3		0.529	0.699

Weight of salt × factor = theoretical weight of reagent necessary to eliminate this salt as a scaling constituent

Parts per million × 0.00833 = 1 lb. per 1000 gal

Grains per U. S. gallon ÷ 7 = 1 lb. per 1000 gal.

Boiler compounds are available in liquid, powdered, or solid form and are introduced into the boiler in various ways. The usual method is to snuff the solution through the injector, feed it to the suction side of the boiler pump by means of a sight-feed lubricator, or pump it from an independent reservoir.

The chief objection to treatment with boiler compound is the accumulation of the scale-forming substances within the boiler itself. This necessitates more frequent blowing off and greater supervision than with outside

treatment. The tendency in the modern plant is to do away with the use of substances within the boiler for reacting chemically with impurities or aiding mechanically in their elimination.

Interior Treatment of Boiler Waters: Railway Age, Nov. 12, 1921, p. 935.
Treating Boiler Scale with Kerosene: Power Plant Engrg., Mar. 1, 1918, p. 212.
The Sphere of Boiler Compound: Power, July 8, 1924, p. 56.

245. Water-softening and Purifying Plants. — Chemical treatment of feedwater outside the boiler is effected in " water-softening " or " purifying " plants. The term " water softener " is ordinarily applied to systems in which the temporary and permanent hardnesses are reduced to a negligible point, and the term " purifying plants " to systems in which some particular impurity or impurities are neutralized or completely removed. In boiler practice the two terms are used synonymously and are applied to all systems of water treatment outside the boiler. Water-softening plants are of two basic types, **precipitation** and **zeolite.** In the former, the reagents are added to the raw water and thoroughly mixed, and the precipitated impurities are removed by sedimentation or filtration. In the latter, chemical action takes place as the raw water gravitates or is forced through a layer of material known as zeolite, which possesses the property of exchanging sodium for calcium and magnesium. Precipitation plants include two types of **cold processes**: the continuous, in which the water flows to the softener in a continuous stream; and the intermittent, in which the water is treated in batches. Where the chemicals are lime or lime and soda, as is usually the case, the plants are sometimes designated as a **lime** or a **lime-soda** plant. The cold-process plant is used chiefly in softening water for locomotives and in large plants where space requirements are not restricted. The hot process is commonly used in plants where exhaust steam is available for heating the water. As chemical reactions are greatly accelerated by heat, the hot-process plant requires less space, lighter foundations, less housing, less piping and fewer fittings than the cold-process plant, and the scale-forming matter is more completely removed and in considerably less time.

The essential elements of the intermittent plant are (1) the **chemical** or **mixing vat** for mixing and dissolving the chemicals, (2) two or more **reaction** or **solution tanks** equipped with stirring devices for mixing the raw water and chemicals, and (3) a **filter.** The essentials of the continuous plant are (1) the **chemical** or **mixing vat,** (2) the **proportioning** device for maintaining the correct ratio of chemicals to water, (3) the **mixing** or **solution tank** for mixing and agitating the water and chemicals, (4) the **precipitation** or **settling tank** for sedimentation and (5) the **filters** for removing any suspended matter which may be carried over from the

settling tank. In either case successful treatment requires a correct ratio of water and reagent, thorough mixing and agitation of both, sufficient time for the completion of the chemical reaction, and complete clarification by sedimentation and filtration. The intermittent and continuous lime-soda processes reduce the hardness of water to an average of about 4 to 5 grains per gallon. Table 78 gives the effect of soda-lime treatment in a specific case.

Figure 382 gives a section through a **Sorge-Cochrane hot process softener** illustrating the continuous hot-process type. An open heater,

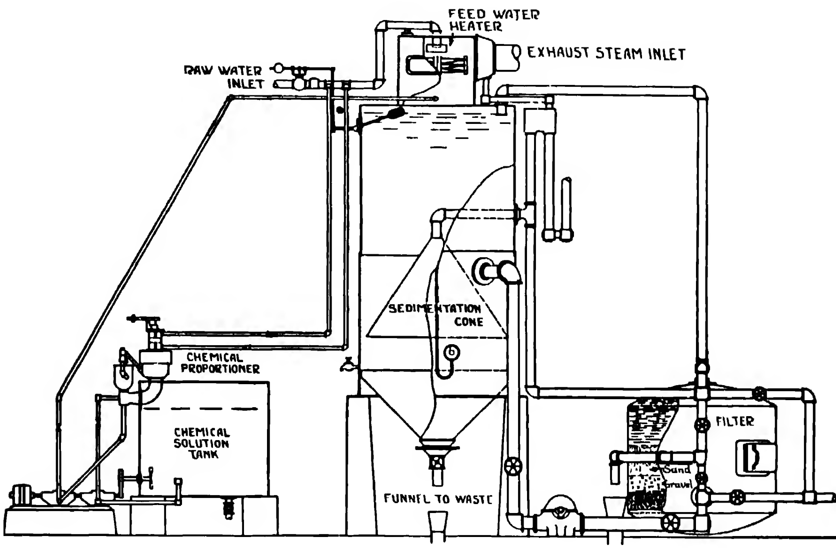


FIG. 382. Sorge-Cochrane Hot-process Softener.

with an oil separator attached for eliminating the greater part of the oil entrained with the exhaust steam, is mounted directly over a chamber in which chemical reaction and sedimentation take place. The raw water is fed into the heater and its temperature is raised to that of the steam. A considerable portion of the dissolved gases is eliminated by this process. The heated water is then mixed with the reagent and falls directly into the sedimentation tank. The precipitated matter is deposited in the conical bottom of the sedimentation chamber, from which it can be washed by the opening of a single valve. In order to eliminate convection currents, the hot water and the softening reagents are delivered at the top and travel slowly to the bottom, from which the clarified water is drawn off by an inverted funnel. The removal of scale-forming matter at high temperatures by sedimentation is so effective that, for many waters and plant

conditions, filtering may be dispensed with. Other conditions demand the use of filters, and in this case a low-pressure sand filter is placed between the sedimentation tank and the boiler-feed pump, the water flowing through the filter by gravity. Modifications of the Sorge-Cochrane softener permit of automatic treatment of part of the water and heating only of the rest, as for example in surface-condenser practice when the condensate is to be heated and the raw makeup water is to be both heated and treated. The hot process reduces the hardness of water to about 3 grains per gallon.

TABLE 78

EFFECT OF SODA-LIME TREATMENT AND FILTRATION

Niagara River — Buffalo, N. Y.

Raw	Grains per U.S. Gal.	Treated	Grains per U.S. Gal.
Volatile and organic matter	trace	Volatile and organic matter ..	trace
Silica .. .	1.85	Silica	0.15
Oxides of iron and alumina	trace	Oxides of iron and alumina .	trace
Calcium carbonate	2.20	Calcium carbonate . . .	1.25
Calcium sulphate	2.11	Magnesium hydrate . . .	0.25
Magnesium carbonate	0.48	Sodium sulphate	2.21
Magnesium chloride	0.05	Sodium chloride	0.80
Magnesium nitrate. . .	1.16	Sodium nitrate	1.31
Sodium chloride...	0.76		
Total solids .. .	8.61	Total solids.	5.97
Suspended matter .. .	0.10		
Free carbonic acid	1.43		
Incrusting substances	7.85	Incrusting substances . .	1.65

Cost of treatment, 0.8 cent per 1000 gallons.

Zeolite water softeners have been in use for several years in laundries, dye establishments, and other industries, but only to a limited extent in boiler plants. Recently, however, the value of this class of softener for treating feedwater has been demonstrated and a great number of plants have put it in service. Zeolites are insoluble hydrous silicates which have the property of exchanging their sodium content for the calcium and magnesium in the water. The exchange does not cease until the sodium is used up, after which the zeolites may be restored to their original efficiency by being soaked in common brine. One of the best known zeolites is marketed under the trade name of **Permutit** and is produced from clay, feldspar, soda ash, and pearl ash. Its composition may be expressed empirically as $2\ SiO_2,\ Al_2O_3,\ Na_2O,\ 6\ H_2O$. Denoting the Permutit radical by the symbol P_e, the softening of water takes place in accordance with the following equations:

$$
\begin{aligned}
Na_2P_e + Ca(HCO_3)_2 &= CaP_e + 2\,NaHCO_3 \\
Na_2P_e + CaSO_4 &= CaP_e + Na_2SO_4 \\
Na_2P_c + Mg(HCO_3)_2 &= MgP_e + Na_2HCO_4 \\
Na_2P_c + MgSO_4 &= MgP_e + Na_2SO_4
\end{aligned}
$$

From the reactions it will be seen that the temporary hardness due to magnesium and calcium bicarbonates is removed with the formation of sodium bicarbonate, and the permanent hardness is removed with the formation of sodium sulphate.

Figure 383 shows a side elevation and sectional end elevation of a typical Permutit softening plant. It will be seen that the raw water is delivered to the top of a closed tank and is caused to percolate successively through a layer of crushed marble, Permutit, and gravel. This filtration effects the necessary purification, and the water leaving the system is of zero hardness, but rich in sodium salts. When the exchange limit is reached, salt water is passed through the zeolite bed and sodium replaces the impurities, which are discharged to the sewer. There is no loss of zeolite material except perhaps that due to attrition.

The advantages and disadvantages of the zeolite system have been summarized by S. T. Powell, as follows:

ADVANTAGES:

1. Treated water from zeolite softeners contains less calcium and magnesium than from any other method with the exception of evaporators.
2. The process requires less attention than any in which chemicals are used.
3. Less space is required to house the apparatus than for a chemical softener.
4. No repumping of water is required.
5. No disposal of sludge is required.
6. Zeolite softening materials will operate with varying hardness of water without changes in methods of operation.
7. Salt, the only agent required for regeneration, is always obtainable and at a reasonable price
8. There is less depreciation than with other types of softeners.
9. No danger exists from deposits of chemicals after treatment.
10. Simplicity of control is fundamental in the method.

DISADVANTAGES:

1. Much higher concentration of soda results than from the lime and soda process.
2. A loss of zeolite material is caused by attrition.
3. The method is not applicable to waters of high hardness, because of the rapid concentration of soda and the high first cost in comparison with lime and soda treatment.
4. It cannot be operated with turbid waters, but must be used in conjunction with filters if the raw water supply contains suspended matter.
5. It cannot be used to soften waters high in iron or manganese unless the water is first treated to remove these constituents.

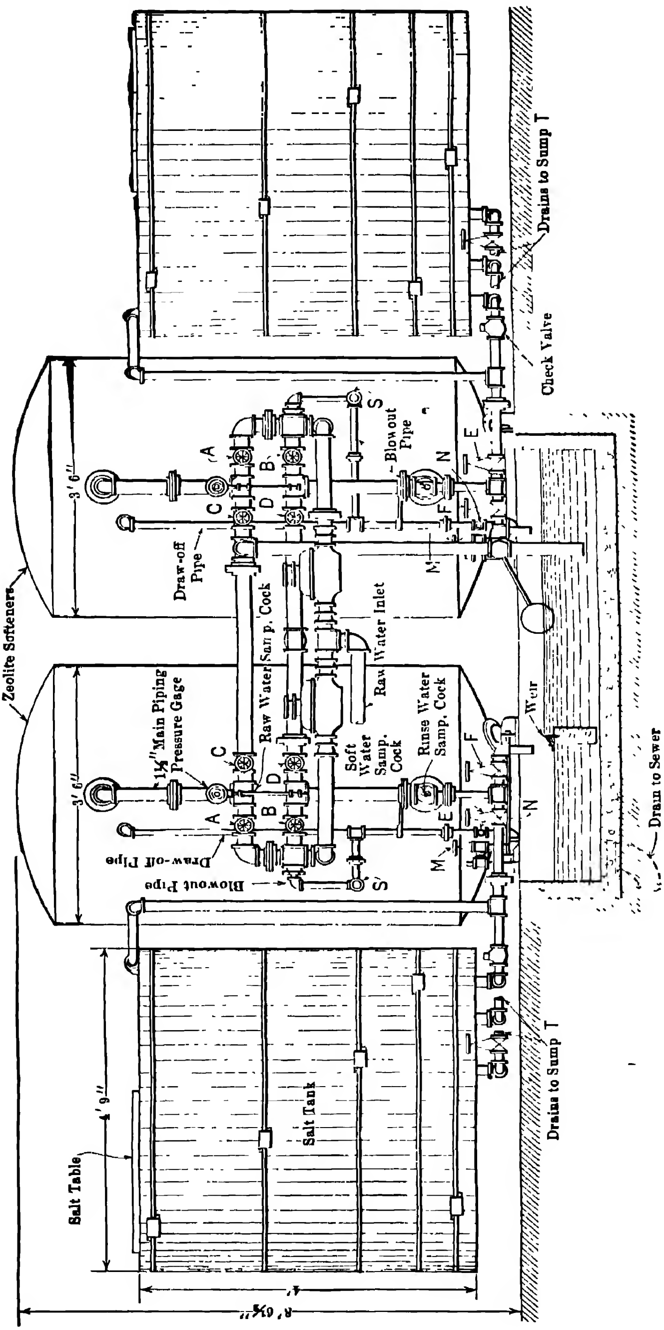


FIG. 383. Typical Permutit Softening Plant.

6. The method is not applicable to the softening of waters which contain acids unless the acid is first neutralized.

Treatment of Feedwater: 1923 Report, Part B, Committee on Prime Movers, N.E.L.A.

Softening Boiler-Feedwater with Zeolites: Power, Sept. 12, 1922, p. 412.

The Relative Merits of Lime-soda and Zeolite Water Softening: Jour. Am. Wat. Wks. Assoc., Vol. 10, No. 4, July, 1923.

246. Protective Coatings. — Various coatings have been applied internally and externally for the purpose of protecting the metal surface from corrosion, and internally only to prevent the adhesion of scale. Graphite, graphite and oil, various organic compounds, galvanizing and carbon paint, have been used for internal surfacing with diversified results. Some plants report that the treatment gave satisfactory results, others that the benefits derived were too short-lived for practical considerations, and a few that the ill effects arising from the use of protective coatings more than offset any noticeable benefit. The application of any coating over a surface which has already become scaled, in the hope of rotting the scale, is not recommended, because the loosened material may lodge in a tube and cause blistering or even failure. Some coatings are greater heat insulators than the scale which they are intended to displace. Internal and external corrosion and scale formation may be prevented by proper feedwater treatment and plant operation, as is evidenced by the performance of many of our modern plants where special attention has been given to the elimination of these troubles.

Distillation. (See paragraph 260.)

247. Degasification. — Internal corrosion due to the presence of dissolved gases in the feedwater may be entirely eliminated by removing the gases from the water before it is fed into the economizer or boiler. There are two distinct processes for effecting this result: (1) **deaëration,** or the liberation of the gases by boiling the water and the subsequent withdrawal of the gases by suitable means, and (2) **deactivation,** or the absorption of the gases by some chemical reagent, such as iron turnings. There are several makes of deaërators and deactivators on the market. Among the former may be mentioned the **Elliott "Contraflo"** and **Cochrane,** and among the latter the **Speller.**

Where there is sufficient exhaust steam to heat the feedwater to 212 deg. fahr., an open heater, with large tray surface or efficient spray nozzles, vented to the vacuum pump or ejector, is capable of reducing the gas content to approximately 0.5 cc. per liter. Where the temperature of the feedwater is less than 210–212 deg. fahr., the water is run into a closed tank, where it is sprayed, or otherwise broken up by spilling over pans, under a vacuum somewhat below that corresponding to the boiling point of the water at its intake temperature. The entrance of the water into

the vacuum chamber causes some of it to flash into vapor (**explosive boiling**), and the vapor, in its violent formation throughout the mass of water, carries with it practically all the dissolved gases. This process of explosive boiling is not necessarily limited to low feed temperatures; in fact, the higher the initial temperature, the better will be the gas elimination. When operating with an open-heater temperature of 210 deg. fahr. and a separator temperature of 188 deg., the Elliott deaërator is guaranteed to remove all but 0.02 cc. of gas per liter.

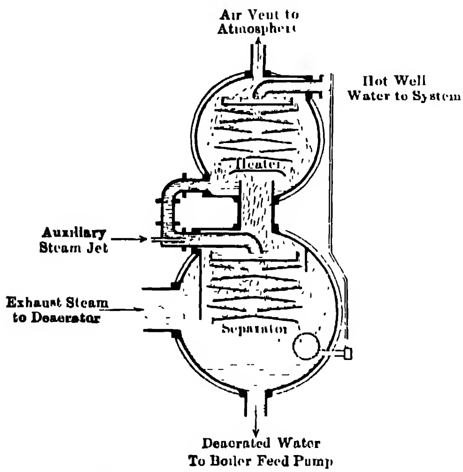


FIG. 384. Cross Section of a Typical Deaërator (Elliott).

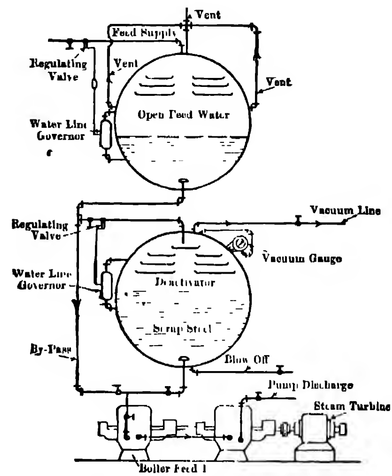


FIG. 385. Combination Deactivation Plant.

In the commercial type of deactivator, the water to be treated is heated as high as possible under the plant conditions, and then run into a closed tank filled with iron turnings or thin perforated plates of any rapid deactivating material, which absorb the oxygen or other free corrosive gases. The higher the temperature, the more rapid is the action of the deactivating material and the smaller can be the apparatus for a given output.

Where the feed temperature is below 200 deg. fahr., and practically complete elimination of both corrosive and other non-condensable gases is desired, a combination of deaërator and deactivator is frequently employed. Figure 385 gives a diagrammatic layout of a combination as designed by the Anti-corrosion Engrg. Corporation. This includes an ordinary vertical open heater from which the water is passed into a tank at lower pressure, but not so low as to boil the water, so that no condenser is required.

The Degasification of Boiler Feedwater: J. R. McDermet, Trans. A.S.M.E., Vol. 44, 1922; Prime Movers Committee, N.E.L.A., 1923, Report, Part B, p. 171.

Field Method for Determining Dissolved Oxygen: Power, Dec. 11, 1923, p. 930.

248. Economy of Preheating Feedwater. — Although a feedwater heater acts to some extent as a purifier, its primary function is that of heating the water. Since the heat content of live steam ranges from 1100 to 1300 B.t.u. per lb. above 32 deg. fahr., 1 per cent less heat is required to evaporate the feedwater into steam for every 11 to 13 deg. that the water is heated. The decrease in fuel consumption, or saving in fuel, due to heating the feedwater will vary with the overall efficiency of the boiler unit. Ordinarily, the temperature of the feedwater does not appreciably affect the overall efficiency, but with some types of boilers, changes in temperature reduce or increase the rate of heat transfer and hence the efficiency.

If H represents the heat content of the boiler steam above 32 deg. fahr., t_0 and t the initial and final temperature of the feedwater, respectively, e the overall efficiency of the boiler unit, then S, the per cent saving in fuel due to preheating, may be expressed

$$S = 100 \frac{(t - t_0)e}{H - (t_0 - 32)}. \tag{236}$$

Example 60. — Steam pressure, 200 lb. gage; superheat, 100 deg. fahr.; initial and final temperature of feedwater, 80 and 210 deg. fahr. respectively; boiler efficiency 75 per cent.

Required saving in fuel due to heating the feedwater.

Solution. — Here H (from steam tables) is 1259, $t_0 = 80$, $t = 210$, $e = 0.75$.

$$S = 100 \frac{(210 - 80)\,0.75}{1259 - (80 - 32)}$$

$$= 8.0 \text{ per cent.}$$

Table 79 based upon equation (236) for 100 per cent boiler efficiency, may be used as a guide in approximating savings due to preheating feedwater.

249. Classification of Feedwater Heaters. — Feedwater heaters may be classified according to the *source* of heat, as

Class	*Source of heat*
1. Exhaust-steam	Exhaust from engines, turbines, etc.
2. Bleeder	Steam bled from intermediate stages of turbines and engines.
3. Jet	Exhaust steam for condenser air ejectors.
4. Gland	Steam used for sealing glands.
5. Flue-gas	Flue gases and waste-heat gases.
6. Live-steam	Steam which has not been partially converted to work.

Or, according to the *method* of heat transmission

Class	*Method*
1. Open	Direct contact of steam and water.
2. Closed	Steam and water separated by metal walls.

TABLE 79

PERCENTAGE OF SAVING FOR EACH DEGREE OF INCREASE IN TEMPERATURE OF FEEDWATER

(Based on Marks & Davis Steam Tables)

Initial Temp. of Feed	Steam Pressure, Lb per Sq. In Gage. (Saturated)										
	0	20	40	60	80	100	120	140	160	180	200
32	0869	.0857	0851	0846	0843	0811	0839	0837	0835	0834	0834
40	0875	0863	0856	0853	0849	0846	0845	0843	.0841	0810	0839
50	0883	0871	0864	0859	0856	0853	0852	0850	0848	.0847	0846
60	0891	.0878	.0871	0867	0864	0861	0859	0857	0855	0854	0853
70	0899	.0886	.0879	0874	.0871	0868	0867	0865	0863	0862	0861
80	0907	0894	0887	0882	.0878	0876	0874	0872	0871	0870	0869
90	0915	0902	0895	0890	0887	0884	0882	0880	0878	0877	0876
100	0924	0910	0903	0898	0895	0892	0890	0888	0886	0885	0884
110	0932	0919	0911	0906	0903	0900	0898	0896	0894	0893	.0892
120	0941	0927	0919	0915	.0911	0908	0906	0904	0902	0901	0900
130	0950	0936	0928	0923	0919	0916	0915	0912	0911	0910	0909
140	0959	0945	0937	0931	0928	0925	0923	0921	0919	0918	0917
150	.0969	.0954	0946	.0940	0987	0933	0931	0930	0928	0927	0926
160	0978	0963	0955	0948	.0946	0942	0940	0938	0936	0935	0934
170	0988	0972	0964	0958	0955	0951	0948	0947	0945	0944	0943
180	0998	0982	0973	0968	0964	0960	0958	0956	0954	0953	0952
190	.1008	0992	0983	0977	0973	0969	.0968	0965	0964	0963	0962
200	1018	1002	0993	0987	0983	0978	0977	0974	0973	0972	0971
210	1029	1012	1003	0997	0993	0989	0987	0984	0983	0982	0981
220	.	1022	1013	1007	1003	0999	0997	0994	0992	0991	.0990
230		1032	1023	1017	1013	1009	1007	1004	1002	1001	1000
240	.	1043	.1034	.1027	1023	1019	1017	1014	1012	.1011	1010
250		1054	1044	1008	1034	1029	1027	1024	.1022	1021	.1020

Multiply the factor in the table corresponding to any given initial temperature of feedwater and boiler pressure by the total rise in feedwater temperature, the product will be the percentage of saving

Heaters may also be classified according to the pressure of the heating steam, as

1. **Vacuum,** or **primary,** in which the pressure is less than atmospheric, as, for example, the exhaust from condensing units and steam bled from the lower stages of a steam turbine. Vacuum heaters are usually of the closed type unless the jet condenser of the house turbine is classed as a heater.

2. **Atmospheric,** or **secondary,** in which the pressure is atmospheric or, literally, that corresponding to the back pressure on the engines and pumps.

3. **Pressure,** in which the pressure is above atmospheric.

Heaters may be still further classified as

a. **Induced,** in which only such steam is admitted as is induced by its condensation. That is, the feedwater condenses the steam. This creates a partial vacuum which draws in more steam.

b. **Through,** in which all the steam is forced through the heater irrespective of condensation.

While all feedwater heaters, condensers, and coolers are heat exchangers, the term **heat exchanger** without qualification is ordinarily applied to small auxiliary appliances where heat is transferred from one liquid to another.

250. Open Heaters.— Figure 386 gives a sectional view of a Cochrane special feed heater and receiver and is a typical example of an open heater. Exhaust steam enters the heater through a fluted oil separator as indicated, and passes out at the top, while the oily drips are automatically drained to waste by a suitable ventilated float. The feedwater enters through an automatic valve and is distributed over a series of copper trays so arranged and constructed that the water is forced

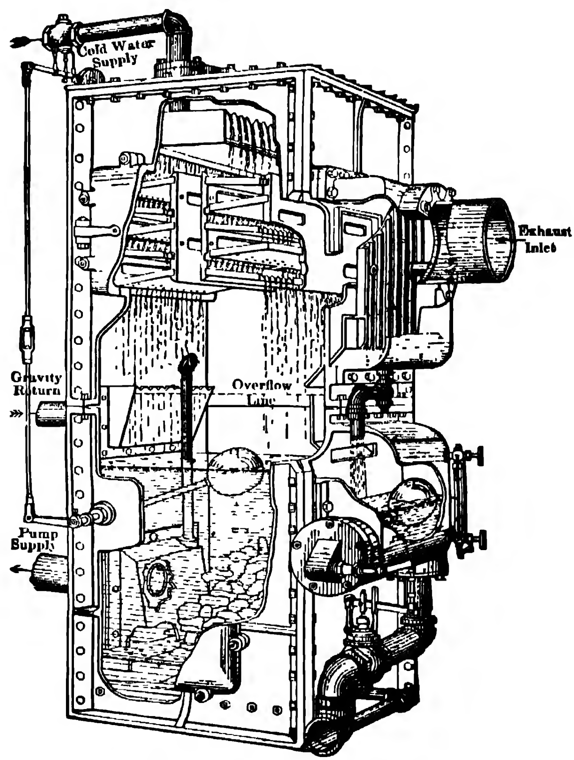


FIG. 386. Cochrane Feedwater Heater.

to fall in a finely divided stream before reaching the reservoir in the bottom. The steam coming in contact with the water particles gives up latent heat and condenses. Some of the scale-forming element is deposited on the surface of the trays, from which it may be removed. The suspended matter is eliminated by a coke filter in the bottom of the chamber, and the floating impurities are decanted by a skimmer

or overflow weir. The particular heater shown in the illustration is especially designed for use in a steam-heating plant; i. e., besides performing all the functions of an open heater, it provides for the reception and heating of the condensation returned to it from the heating system.

Figure 387 shows a section through a **Hoppes** open heater, illustrating the "pan" type. Exhaust steam enters at H, passes through oil filter O, and completely surrounds pans T, T. The feedwater enters at B, and

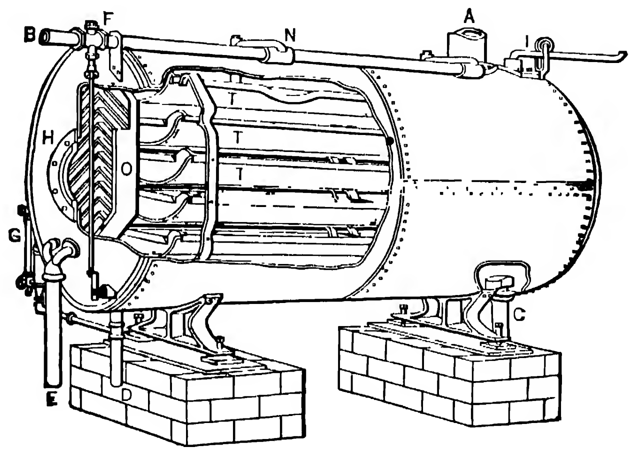


FIG. 387. Typical Open Heater, Horizontal. (Hoppes.)

the rate of flow is regulated by valve F, which is controlled by a suitable float in the lower part of the chamber. The water, in flowing over the sides and bottoms of the pans, comes into direct contact with the steam.

251. Combined Open Heater and Chemical Purifier. — Combined feedwater heaters and chemical purifiers are finding increased favor with some engineers in districts where the feedwater is particularly bad and space limitations preclude the use of water-softening plants. When properly proportioned, the purification is highly satisfactory, but as a general rule the equipment is too small and sufficient time is not given for efficient purification.

252. Heat Exchange in Open Heaters. — The various factors entering into the heat exchange between the steam and water in an open heater may be correlated as follows:

Let H, q_0 and q represent, respectively, the heat content per lb. of steam, inlet water and outlet water, B.t.u.; t_0 and t the initial and final temperature of the water, deg. fahr.; and W_1 and W_2 the weight of steam condensed and water heated, lb. per hr.

Then, $W_1 (H - q)$ = heat given up by the steam, and $W_2 (q - q_o)$ = heat absorbed by the water.

Therefore, neglecting losses,

$$W_1 (H - q) = W_2 (q - q_o)$$

which for all practical purposes may be written

$$W_1 (H - t + 32) = W_2 (t - t_o) \tag{237}$$

This equation is applicable to all phases of *open* and *closed* heater practice. Attention is called to the fact that the maximum value of t can never exceed that of the steam used for heating.

Example 61. — A 1000-hp. non-condensing uniflow engine used 24.5 lb. of saturated steam per i.hp-hr.; initial pressure 150 lb. abs.; back pressure 0 lb. gage; temperature of water supply 62 deg. fahr. Required the percentage of the engine steam supply which must be used for heating the feedwater to the maximum obtainable.

Solution. — The maximum temperature possible with steam at atmospheric pressure is 212 deg. fahr.; i. e., $t = 212$. H may be calculated from equation (146) assuming a loss of 1 per cent, thus:

$$H = 1193.4 - 0.01 \times 1193.4 - 2547/25 = 1079.7$$

In open-heater practice the total weight of hot water available is that of the condensate plus that of the cold water supplied. The total weight required in this problem, assuming no losses, is $24.5 \times 1000 = 24{,}500$ lb. per hr.; therefore, W_2, the weight of cold water to be supplied $= 24{,}500 - W_1$. Substituting $W_2 = 24{,}500 - W_1$, $H = 1079.7$, $t = 212$, and $t_o = 62$ in equation (237) and solving for W_1 we have

$$W_1 (1079.7 - 212 + 32) = (24{,}500 - W_1) (212 - 62)$$

from which

$$W_1 = 3500, \text{ approximately,}$$

and

$$W_1/24{,}500 = 3500/24{,}500 = 0.143 \text{ or } 14.3 \text{ per cent.}$$

Example 62. — 20,000 lb. of steam per hr. are bled from the 17 lb. per sq. in. abs. stage of a steam turbine to an open heater. If the turbine water rate is 21 lb. per kw-hr. at the 17 lb. stage, required the temperature to which the feedwater can be heated if the total weight of feedwater is 300,000 lb. per hr. Initial pressure 300 lb. abs., superheat 200 deg. fahr., temperature of condensate 92 deg. fahr.

Solution. — W_2, the weight of condensate to be heated, $= 300{,}000 - 20{,}000 = 280{,}000$ lb. per hr.

From equation (146), assuming the turbine and generator efficiency to be 95 per cent,

$$H = 1319 - 3415/(21 \times 0.95) = 1148.$$

Substituting $H = 1148$, $t_o = 92$ and $W_2 = 290{,}000$ in equation (237) and solving for t we have

$$20{,}000\ (1148 - t + 32) = 280{,}000\ (t - 92)$$
$$t = 165 \text{ deg. fahr., approximately.}$$

253. Pan Surface Required in Open Feedwater Heaters. — Pan or tray surface required varies according to the quality of the water with regard to both scale-making material and mud, but may be approximated by the formula

$$\text{Pan surface, sq. ft.} = \frac{\text{Pounds of water heated per hour}}{c}. \qquad (238)$$

	Vertical Type	Horizontal Type
For very muddy water, c	118	110
Slightly muddy water, c	166	155
For clean water, c.	500	400

254. Size of Shell, Open Heaters. — General proportions of open heaters vary considerably on account of the different arrangements of pans or trays, filters, and oil-extracting devices. A fair idea of the size of shell required may be obtained by the formulas

$$\text{Area of shell} = \frac{\text{horsepower}}{a \times \text{length in feet}} \qquad (239)$$

$$\text{Length of shell} = \frac{\text{horsepower}}{a \times \text{area in square feet}} \qquad (240)$$

$a = 2.15$ for very muddy water,
$a = 6$ for slightly muddy water,
$a = 8$ for clean water.

The horsepower in this case is obtained by dividing the weight of water heated per hour by the steam consumption of the engine per horsepower per hour.

Pans containing 2.5 sq. ft. and less are usually made round, and larger sizes rectangular in plan. When circumstances will permit, it is better to have not more than six pans in any one tier, since it is advisable to proportion the pans so as to obtain as low a velocity over each as practicable.

Distance between trays or pans is seldom less than one-tenth the width for rectangular, and one-fourth the diameter for round pans. Volume of storage and settling chamber in horizontal heaters varies from 0.25 for good quality of water to 0.4 of the volume of the shell for muddy water, 0.33 being about the average. In the vertical type, the settling chamber

represents respectively 0.4 and 0.6 the volume of the shell with clear and muddy water. Filters occupy from 10 to 15 per cent of the volume of the shell in the horizontal type and from 15 to 20 per cent in the vertical type, the smaller percentage corresponding to clear water and the larger to muddy water or water containing a considerable quantity of impurities.

255. Closed Heaters. — Closed steam heaters bear the same relationship to open heaters as do surface condensers to jet condensers; in fact, all steam heaters are condensers. In all surface condensers, except those of the water-works type, the cooling water passes through the tubes and the steam passes across or around the tubes, while in the majority of closed water heaters the reverse is true. In surface condensers, the tubes are invariably straight; but in closed heaters, because of the higher temperatures, the tubes are frequently bent, coiled, or corrugated to provide for the excessive expansion. Closed heaters operate with either **parallel** or **counterflow,** and the water passes directly through a single nest of tubes (**single flow**) or back and forth through a series of nests (**multi-flow**). Occasionally, where scale-free water is available, the water is forced across the heating surface in a thin sheet or film (**film heaters**).

Figure 388 shows a section through a multi-flow straight-tube closed heater illustrating the type most commonly found in power plant practice.

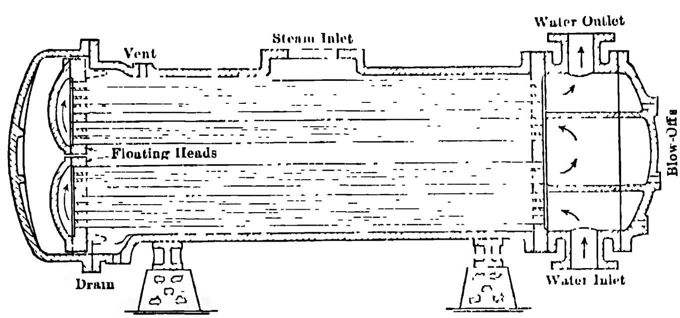


FIG. 388. Typical Closed Heater with Floating Heads. (Alberger.)

The heads are of cast iron and the shell of cast iron or sheet steel. The tubes are rolled into a fixed head at the inlet end and into a loose or **floating head** at the other end, thus providing for contraction or expansion. Removable covers afford easy access to the tubes without breaking steam or water connections. Condensate is removed from the bottom of the steam chamber by a suitable drip. Both the fixed and floating heads are baffled so as to cause the water to pass back and forth through different nests of tubes. This increase of length of water travel permits of higher velocity of flow with corresponding increase in rate of heat transmission.

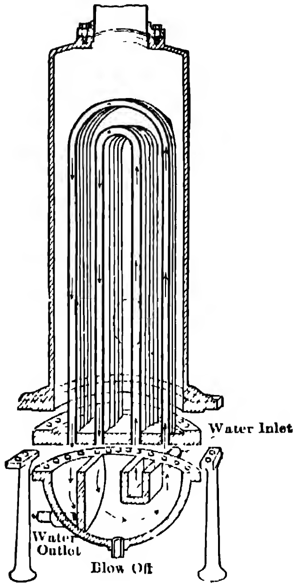


FIG. 389. Typical U-tube Closed Heater. (Berryman Type.)

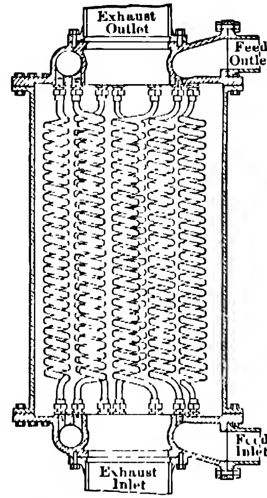


FIG. 390. Typical Multi-tube Coiled Closed Heater. (Reilly.)

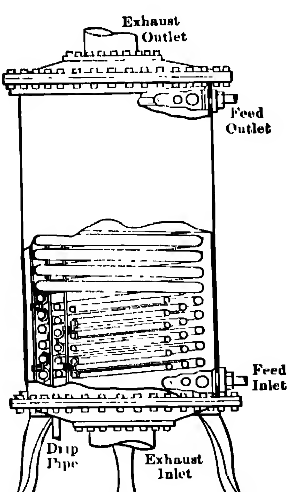


FIG. 391. Typical Coil Heater. (National.)

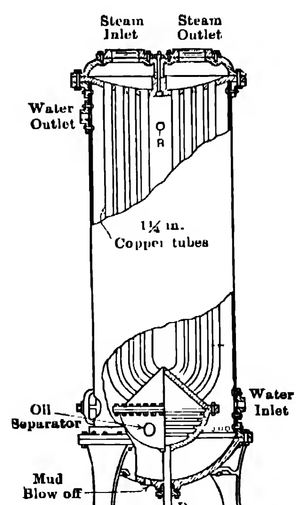


FIG. 392. Typical Steam-tube Feedwater Heater. (Otis.)

Figure 389 shows a section through a closed heater in which expansion is absorbed by bending the tubes as indicated. In Fig. 390 and Fig. 391 the tubes are coiled, giving a long water travel and at the same time providing for expansion.

Steam-tube heaters are sometimes employed because the scale adheres to the outside of the tubes instead of the inner surfaces. Where the scale is of such a nature that it can be readily loosened by any simple treatment, the accumulation can be easily removed by washing through the various openings installed for this purpose. Figure 392 shows a section through a heater of this type.

The heating element in a **film** heater consists usually of two spirally corrugated tubes, one within the other, the water path being the small annular clearances between the two. Thus the water is directed in a spiral path due to the corrugations, and for a given velocity the particles of water come more often in contact with the heating surface than in plain tubes, because they are contained within an annular space whose perimeter is large in comparison with its area. This type of heater, though highly efficient in heat transmission, necessitates the use of comparatively pure water and is not commonly used for heating raw water.

256. Heat Transmission in Closed Heaters. — Since the closed heater is practically the same in principle as a surface condenser, the laws of heat transmission are practically identical in both cases, at least for maximum water temperatures under 180 deg. fahr. Above this temperature, the occluded gases appear to have a marked influence on the amount of heat transferred. (Consult "The Laws of Heat Transfer," *Engrg.*, Vol. 116, July 6 to Aug. 24, 1923.) Because of the liberal factors allowed in practice, it is sufficiently accurate for most engineering designs to assume that the fundamental laws for water heaters and condensers are the same. Increasing the velocity of the fluid which is to be heated in passing through the heater increases the rate of heat transmission and thereby renders the heating surface more effective. In order to employ moderately high velocities and at the same time allow sufficient time in which to raise the temperature to a maximum, the passages through the heater should be as long as practicable and of small cross-sectional area. Other things being equal, a heater containing a large number of passages of small cross-sectional area is more efficient than one containing a small number of large passages. It is important to proportion the heater according to the amount of fluid to be heated and the maximum temperature to which the fluid must be raised. In designing a heater, then, the maximum temperature to which the fluid is to be raised and the coefficient of heat transfer are assumed and the amount of heating surface is calculated from equation (241) or (242).

Although recent experiment shows that the amount of heat transmitted through the heating surface is proportional to some power of the mean temperature difference, the value of the exponent is not far from unity (0.8 to 0.9) and it may be safely taken as such, particularly in view of the liberal factor allowed in the assumed value of the coefficient of heat transfer, U. With this assumption, the extent of heating surface may be calculated from the following modification of equation (209).

$$S = cw\ (t_2 - t_o) \div Ud \tag{241}$$

in which

S = total tube heating surface, sq. ft.,
c = mean specific heat of the fluid to be heated; for water this may be taken as 1.0,
w = weight of fluid heated per hr., lb.,
t_2 = final temperature of the fluid, deg. fahr.,
t_o = initial temperature of the fluid, deg. fahr.,
U = mean coefficient of heat transfer for the entire surface, B.t.u. per sq. ft. per deg. difference in temperature per hr.,
d = mean temperature difference between the steam and that of the fluid to be heated.

For ordinary practice, where the various influencing factors are not well established, it is sufficiently accurate to take the arithmetic mean as given in equation (219). Heater manufacturers, however, usually base their calculation on the logarithmic mean as given in equation (218).

Substituting the logarithmic value of d in equation (241) and reducing we have

$$S = \frac{cw}{U} \log_e \frac{t_s - t_o}{t_s - t_2}. \tag{242}$$

For a given extent of heating surface S, the temperature difference between that of the steam and the feedwater leaving the heater may be calculated by solving equation (242) for $t_s - t_2$, thus

$$(t_s - t_o) - (t_s - t_2) = e^n \tag{243}$$

in which

e = base of the Naperian logarithm
= 2.718,
$n = SU/cw$; for water $n = SU/w$.

By taking different extents of area, S, and solving for the corresponding

values of $t_s - t_2$, the temperature gradient for a given heater may be obtained as illustrated in Fig. 393.[1]

From equation (241) it will be seen that the extent of heating surface depends upon the weight and specific heat of the fluid to be heated, the temperature of the steam, the desired final temperature of the fluid, and the value of coefficient of heat transfer, U.

Since the extent of heating surfaces increases rapidly as t_2 approaches t_s, and becomes infinity for $t_2 = t_s$, it is desirable to limit t_2 to some practical figure. An average maximum of t_2 for feedwater heaters $= t_s - 4$.

The coefficient of heat transfer in tubular feedwater heaters varies within wide limits, depending upon the type of heater and the conditions of operation, and ranges from $U = 150$ in steel tube heaters with low water velocities to 1000 or more in the film type of corrugated brass tube heaters with water velocity of 7 ft. per second. With superheated steam the amount of heat transferred through the tubes will be practically the same as with saturated steam, the pressure being the same in each case. This is due to the fact that the outer surface of the tube cannot rise under practical conditions above the saturation temperature of the steam, regardless of the amount of superheat. Therefore, the same value of U may be taken for both saturated and superheated steam, since the temperature difference between the circulating water and the outer surface of the tube will be the same in each case. (Consult "Superheated Steam Used Directly in Closed Heaters," by B. C. Sprague, *Power*, Jan. 29, 1924, p. 161.) In large central stations operating with highly superheated steam, the exhaust from the steam-driven auxiliaries and from the high-pressure bleeding stage of the turbine is frequently superheated. In such installations the exhaust or bled steam is sometimes "desuperheated" before entering the heater. In practice a liberal factor is allowed for possible heat reduction due to the presence of air and the accumulation of oil, scale, or other deposits on the tube surfaces. The "average" values

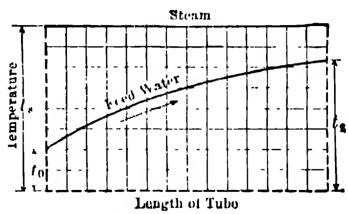


FIG. 393. Temperature Gradient in Feedwater Heater Tube.

[1] For greater accuracy in predicting the temperature gradient, D. K. Dean (Indus. and Engrg. Chem., May, 1924, p. 483) offers the following modification of equation (242)

$$S = \frac{w}{b\,(t_s + a)} \log_e \frac{(t_s - t_0)\,(t_2 + a)}{(t_s - t_2)\,(t_0 + a)},$$

in which a and b are constants for a given design and set of operating conditions. Other notations as in equation (242). For the method of determining a and b consult the reference in question.

TABLE 80

SQUARE FEET OF HEATING SURFACE REQUIRED TO HEAT 1000 POUNDS OF WATER PER HOUR

$U = 350$

Vacuum Heaters between Engine and Condenser

Initial Temperature of Feedwater t_o	24 in. Vacuum, Temperature 141 deg. fahr.						25 in. Vacuum, t_s = 134 deg. fahr.			
	Final Temperature of Feedwater									
	105	110	115	120	125	130	110	115	120	125
40	2.93	3.36	3.86	4.50	5.22	6.29	3.93	4.65	5.58	7.01
50	2.64	3.29	3.57	4.15	4.93	6.01	3.65	4.36	5.28	6.65
60	2.29	2.93	3.22	3.86	4.58	5.65	3.29	4.01	4.93	6.29
70	1.93	2.50	2.86	3.43	4.22	5.29	3.07	3.58	4.57	5.86
80	1.50	2.07	2.43	3.01	3.72	4.86	2.36	3.07	4.01	5.36

t_o	26 in. Vac. t_s = 125°				27 in. Vac. t_s = 114°			28 in. Vac. t_s = 100°		
	Final Temperature of the Feedwater									
	105	110	115	120	90	100	105	70	80	90
40	4.43	4.93	6.07	8.18	3.22	4.72	6.01	1.93	3.14	5.08
50	3.92	4.57	5.72	7.73	2.79	4.36	5.58	1.43	2.57	4.57
60	3.36	4.15	5.36	7.51	2.29	3.86	5.08	.78	1.93	3.93
70	2.86	3.65	4.79	6.86	1.71	3.22	4.50		1.14	3.14
80	2.29	3.07	4.28	6.28	.86	2.21	3.79			1.93

Atmospheric Heaters

Initial Temperature of Feedwater, t_o	Atmospheric Pressure, Temp. 212 deg. fahr.					
	Final Temperature of Feedwater					
	192	196	200	204	208	210
40	6.01	6.65	7.58	8.73	10.72	12.74
50	5.94	6.58	7.44	8.58	10.51	12.51
60	5.79	6.44	7.15	8.36	10.38	12.30
70	5.58	6.22	7.01	8.23	10.15	12.15
80	5.37	6.07	6.87	8.00	9.94	11.85
90	5.15	5.79	6.57	7.80	9.80	11.79
100	4.93	5.51	6.36	7.51	9.52	11.50
105	4.79	5.46	6.22	7.37	9.37	11.37
110	4.65	5.28	6.15	7.22	9.23	11.15
115	4.50	5.15	5.93	7.08	9.08	11.07
120	4.36	4.93	5.79	6.93	8.95	10.94
125	4.15	4.79	5.64	6.79	8.80	10.80
130	3.86	4.65	5.44	6.57	8.65	10.58

For any other value of U, divide the tabular value by 350 and multiply by the new value of U.

of U in Table 81 are very conservative, and a heater based upon these figures will operate with the predetermined high water temperatures for a long time without cleaning. With scale-free water and steam, and high water velocities, the higher values are frequently used. For steam coils submerged in water and from which the condensation is withdrawn as rapidly as it is formed, the value of U in Table 89 appears to give satisfactory results.

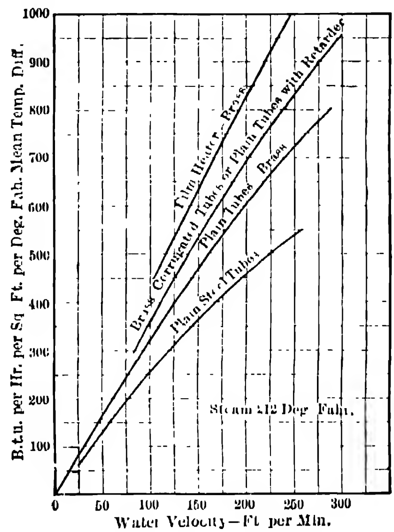


FIG. 394. Coefficient of Heat Transfer (For General Design.)

For steam-air heaters, under practically atmospheric conditions, the mean value of U for plain brass tubes varies from 1/2 to 25, depending primarily upon the initial condition of the steam and the velocity of the air through the passage, the lower value for steam at 27-in. vacuum and air velocity of 10 ft. per sec., and the higher value for steam at 100 lb. gage and air velocity of 30 ft. per sec. For extended surfaces, as in the "U-fin" construction, Fig. 380, the heat transfer is considerably higher than for plain tubes.

For **steam-oil** heaters, the number of variables is so great that specific data must be had from the manufacturer. The values in Table 83 are for a specific type of heater and set of operating conditions and are of interest merely in showing the variation in the value of U with the velocity of the oil.

Example 63. — The exhaust from a 200-hp. single non-condensing engine is to be used for heating water in a closed heater. If the water rate of the engine is 30 lb. per i.hp-hr., initial pressure 115 lb. abs., dry steam at admission, required the amount of water which can be heated by the exhaust from an initial temperature of 58 deg. to a final temperature of 208 deg. fahr., and the extent of heating surface necessary to effect this result, assuming $U = 250$.

Solution. — From equation (146), assuming $H_r = 0.01\ H_i$, we find

$$H = 1188.8 - 2547/30 - 0.01 \times 1188.8 = 1092 \text{ B.t.u.}$$

Substituting $H = 1092$, $t_o = 58$ and $t = 208$ in equation (237) and solving,

$$200 \times 30\ (1092 - 208 + 32) = W_2\ (208 - 58),$$

from which $W_2 = 36{,}640$ lb. of water heated per hr.

Substituting $W = W_2 = 36{,}640$, $t_s = 212$, $t_o = 58$, $t_2 = 208$ and $U = 250$ in equation (242) and solving, we have

$$S = \frac{36{,}640}{250} \log_e \frac{212 - 58}{212 - 208} = 535 \text{ sq. ft.}$$

Example 64. — Determine the length of 3/4 in. (O.D.), 1/16 in. thick brass tubes in a closed heater designed to heat water from 60 to 196 deg. fahr., steam temperature 212 deg. fahr., water velocity 2 ft. per sec., $U = 400$.

Solution. — $S = \pi dl/12 = 3.14\, dl/12 = 0.197\, l.$

l = water travel or total length of pass,

$$W = \frac{2 \times 3600 \times \pi d^2 \delta}{144 \times 4} = \frac{7200 \times 3.14 \times (5/8)^2 \times 62.4}{144 \times 4} = 957 \text{ lb. per hr.}$$

Substituting these values in equation (241),

$$0.197\, l = \frac{957}{400} \log_e \frac{212 - 60}{212 - 196}$$

From which $l = 27.3$ ft. approx.

Example 65. — A 200-sq.-ft. closed heater is rated at 40,000 lb. of water per hour, initial temperature 60 deg. fahr., steam temperature 212 deg. fahr., $U = 300$. Required the final temperature of the water.

Solution. — Here $e = 2.718$, $S = 200$, $U = 300$, $w = 40{,}000$, $n = SU/w = 200 \times 300/40{,}000 = 1.5$, $t_s = 212$, $t_o = 60$. Substituting these values in equation (243) and solving, we have

$$(212 - 60) \div (212 - t_2) = 2.718^{1.5}$$
$$t_2 = 178.1 \text{ deg. fahr.}$$

TABLE 81

HEAT TRANSMISSION IN CLOSED FEEDWATER HEATERS

(Based on Commercial Designs)

Type of Heater	Coefficient of Heat Transfer, U	
	Range	Average*
Single-flow, plain brass tubes	150–500	200
Single-flow, corrugated brass tubes	250–600	300
Single-flow, steel tubes	125–250	150
Spiral coils, plain brass tubes	250–800	350
Multi-flow, plain brass tubes	250–800	350
Multi-flow, corrugated brass tubes	350–900	400
Plain brass tubes with retarders	350–1000	450
Film heater with corrugated tubes	500–1200	600

* Because of the many variables entering into the problem of heat transfer, these values are of academic interest only. Specific data should be had from the manufacturer.

TABLE 82

HEAT TRANSFER — SUBMERGED STEAM COILS

Mean Temperature Difference	Coefficient of Heat Transfer, U		
	Iron	Brass	Copper
50	100	200	220
100	175	275	300
150	200	375	400
200	225	450	475

TABLE 83

HEAT TRANSFER — STEAM TO FUEL-OIL, $\frac{5}{8}$ IN 15 GAGE STEEL TUBES

Velocity of Oil Ft. per Sec.	U	Velocity of Oil Ft. per Sec.	U	Velocity of Oil Ft. per Sec.	U
0 2	20	1 0	100	1 8	143
0 4	55	1 2	112	2 0	153
0 6	71	1 4	123	2 2	162
0 8	86	1 6	133	2 4	172

Economic Features of Heat-exchanger Design. Mech. Engrg., Dec. '24, p. 891.

257. Open vs. Closed Heaters. — Open and closed heaters have their respective advantages, and a careful study of the various influencing conditions is necessary for an intelligent choice. The following parallel comparison brings out a few of the distinguishing features:

OPEN HEATER	CLOSED HEATER
Efficiency	
With sufficient exhaust steam for heating the feedwater may reach the same temperature as the steam.	The maximum temperature of the feedwater will always be 2 degrees or more lower than the temperature of the steam
Scale and oil do not affect the heat transmission.	Scale and oil deposit on the tubes and the heat transmission is lowered.
Pressures	
It is not ordinarily subjected to much more than atmospheric pressure.	The water pressure is slightly greater than that in the boiler when placed on the pressure side of the pump as is customary.
Safety	
Sticking of the back pressure valve may cause it to "blow up" if provision is not made for such an emergency.	It will safely withstand any pressure likely to occur.

	Purification
Since the exhaust steam and feedwater mingle, provision must be made for removing the oil from the steam.	Oil does not come in contact with the feedwater.
Scale and other impurities precipitated in the heater are readily removed.	Scale is removed with difficulty.
Dissolved gases are removed if heater is properly ventilated.	Does not remove dissolved gases, unless vented to lower pressures.
	Location
Must always be placed above the pump suction and on the suction side	May be placed anywhere on the pressure side of the pump.
	Pumps
With supply under suction, two pumps are necessary and one must handle hot water.	One cold-water pump is necessary.
	Adaptability
Particularly adaptable for heating systems where it is desired to pipe the "returns" direct to heater.	Vacuum or primary heaters are usually of this type.
	Adaptable to stage bleeding.

258. Arrangement of Heaters. — Figure 395 shows a typical installation of an open heater connected as a " through " heater. This arrangement was common in the older designs of non-condensing plants but has been

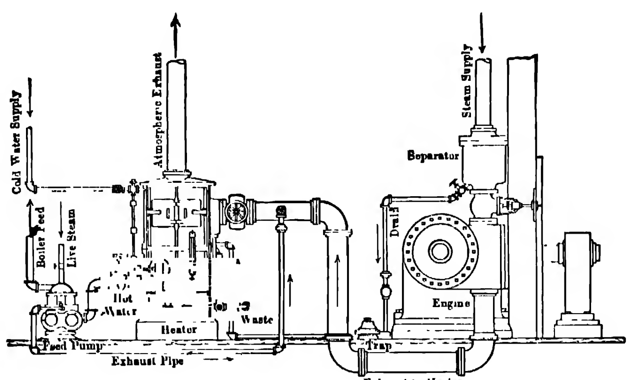


FIG. 395. Typical "Through" Heater.

practically superseded by the " induced " connection as shown in Fig. 397. It is evident that *all* the steam must pass *through* the heater. Now, 1 lb. of exhaust steam in condensing gives up approximately 1000 B.t.u. Hence, if the initial temperature of the feedwater is 50 degrees and the final temperature 210, the engine furnishes $1000/(210 - 50) = 6.26$, say, six times the quantity necessary for heating the feedwater to a maxi-

mum. Therefore, the area of the pipe supplying the heater with steam need be but one-sixth that of the main exhaust. With the heater connected as in Fig. 395, the connections must necessarily be the same size as the exhaust pipe.

With this arrangement the heater cannot be "cut out" while the engine is in operation, and hence it is not adapted for plants working continuously. For the purpose of cutting out a heater while the plant is in operation, a through heater may be by-passed as in Fig. 396. Advantage may be taken here of the permissible reduction in the size of pipes and fittings; i. e., valves, etc., at C and D need be but one-half the size of those at A. This reduction in size may prove to be a considerable item in large installations.

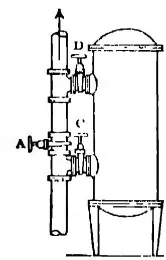


FIG. 396. By-passed Heater.

Figure 397 shows a typical installation of an "induced" heater in a non-condensing plant, which is representative of current practice. In the arrangement in Fig. 397 the number of fittings is reduced to a minimum and the heater may be readily cut out. Since induced heaters are apt to become air bound, a vapor pipe or vent connected to a trap is inserted in the top of the heater as shown. Figure 398 shows a typical installation of an open

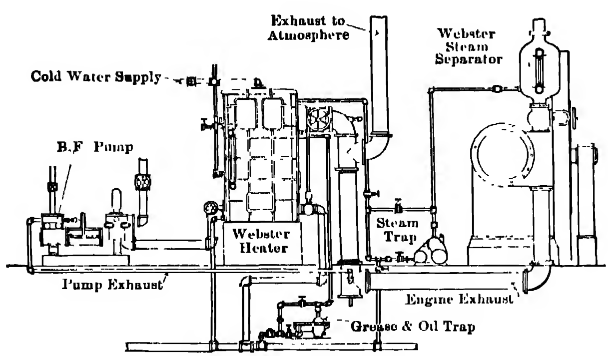


FIG. 397. Typical "Induced" Heater.

heater in a condensing plant, in which the exhaust from the auxiliaries is used for feedwater and house heating and the deficiency is made up by bleeding the turbine.

Other arrangements of heaters will be found in connection with the discussion on the station heat balance, see paragraph 265.

259. Live-steam Heaters and Purifiers. — The function of a live-steam heater and purifier receiving steam at boiler pressure is primarily that of

purification. Live-steam heaters are seldom installed even though the feedwater contains scale-forming elements such as sulphates of lime and magnesia. These, as previously stated, are not entirely precipitated until a

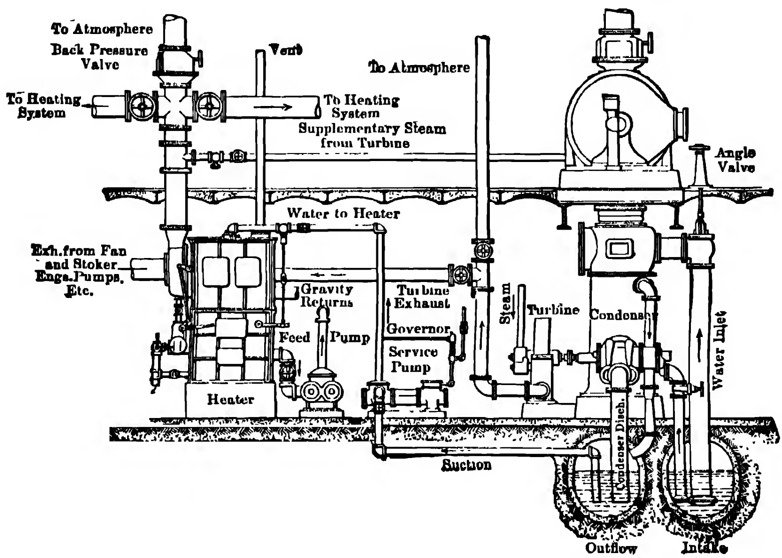


FIG. 398. Open Heater in a Condensing Plant.

temperature of approximately 500 deg. fahr. is reached; hence no amount of heating with exhaust steam at atmospheric pressure will thoroughly purify feedwater containing these elements. If properly vented, all of the dissolved gases may be removed in this manner.

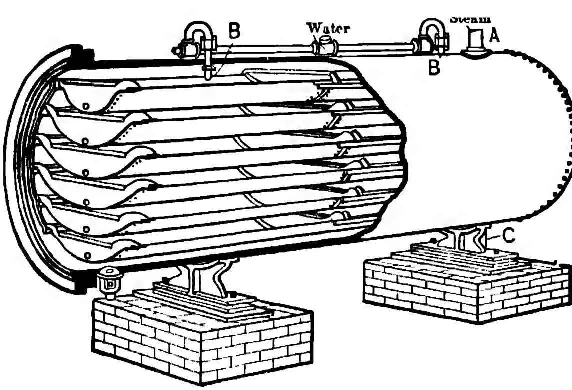


FIG. 399. Hoppes Live-steam Purifier.

Figure 399 shows a section through a Hoppes live-steam purifier. Since the purifier is subjected to full boiler pressure, the shell and heads are constructed of steel. Within the shell are a number of trough-shaped pans or trays placed one above another and supported on steel angle ways. Steam from the boiler enters the chamber at *A* and comes in contact with

the feedwater and condenses. The water on entering the heater at B is fed into the top pan and, overflowing the edges, follows the under side of the pan to the center and drops into the pan below. It flows over each successive pan in the same manner until it reaches the chamber at the bottom, whence it gravitates to the boiler through pipe C. As the steam inclosed in the shell comes in contact with the thin film of water, the solids held in solution are separated and adhere to the bottom of the pans in the same manner that stalactites form on the roofs of natural caves. Authentic tests show that live-steam heaters may increase the boiler efficiency to a

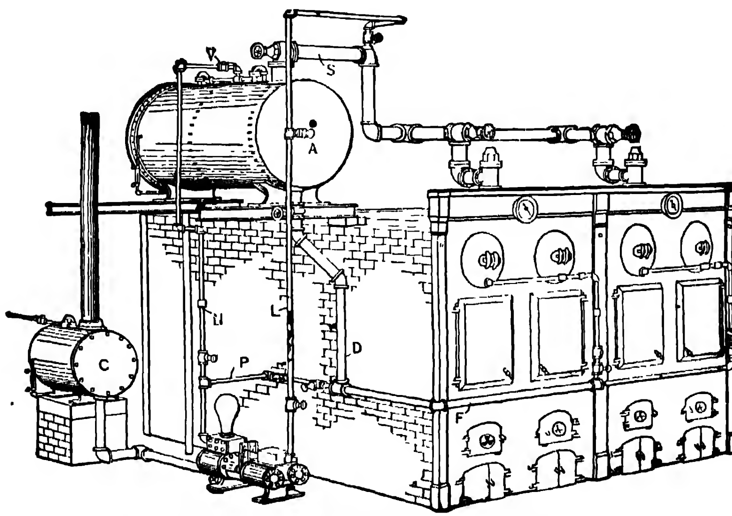


FIG. 400. Typical Installation of a "Live-steam" Purifier.

slight extent, but in most instances there is a slight loss due to the heat given off the heater shell to the surroundings. (See *Power*, Feb. 21, 1911, p. 295.) The purifier should be set in such a position as will bring the bottom of the shell 2 ft. or more above the water level of the boilers, as in Fig. 400. N is the feed pipe from pump to purifier and should be provided with a check valve. D is the gravity pipe through which the purified water flows to the boiler. This pipe should be carried below the water level of the boilers and all branch pipes should be taken off below the water line. Pipe L leads from top of pipe S to pump or other steam-using device. This is necessary in order that air and other non-condensable gases liberated from the water may be removed from the purifier, which would otherwise become air-bound. In the illustration, the feed pump takes its supply from an exhaust-steam heater C. The purifier is provided with a suitable by-pass so that the water may be fed directly to the boiler when necessary.

Heaters receiving steam bled from the high-pressure stages of steam turbines are not classified as live-steam heaters.

260. Distillation. — Sea water, which contains a high concentration of sodium compounds, cannot be purified by any practical chemical treatment, and the only means of removing the dissolved solids is by evaporation. For this reason marine steam plants are invariably equipped with distilling plants. Certain inland waters also contain a large percentage of sodium salts and evaporation is the only possible treatment. Properly deaërated distilled water is practically free from all impurities and permits of high rates of evaporation with minimum cost of upkeep, so that its use for feed purposes may be advisable even where raw water has a low sodium content. This is particularly true in the large central station with its heavy peak loads and high overall rate of driving. While distillation is productive of the purest water, it is not necessarily the most economical method of treatment, since the gain due to the use of pure distillate may be more than offset by the increased investment and operating cost of the evaporating plant.

The most common method of obtaining distilled water is by means of evaporators using steam as a heating medium. The steam supplied may be at any pressure and temperature, provided the condenser or its equivalent receiving the vapor can maintain a pressure below that corresponding to the final temperature of the distillate. Evaporator systems may be classified as **pressure systems,** in which the steam is supplied under pressure considerably above atmospheric, and **vacuum systems,** in which the pressure is at or below atmospheric. Evaporators may be further classified as **low-heat-level,** in which evaporation takes place at low temperature and corresponding pressure, and **high-heat-level,** in which evaporation takes place at high temperature and corresponding pressure. The low-heat-level evaporator usually exhausts to an open heater at a pressure a little above atmospheric, while the high-heat-level apparatus exhausts to a special heater or surface condenser through which the feedwater passes after leaving the main open heater so that the temperature of the vapor will be well above 212 deg. fahr., and at a pressure well above atmospheric. The evaporators of a low-heat-level distilling plant may be operated with boiler steam at full or reduced pressure or with auxiliary exhaust steam at a pressure as low as 2 lb. gage. If operated with low-pressure exhaust steam, the vapor produced will have to be maintained at a vacuum. High-heat-level evaporators are usually operated on full boiler-steam pressure and the resulting high temperature vapor is condensed in a high-heat-level condensing system using boiler feed at approximately 210 deg. fahr. for cooling water. With this system the boiler feedwater reabsorbs all the heat given up in the evaporator

coils with the exception of the small amount lost through radiation and evaporator " blow."

Evaporators may also be classified according to the means of effecting evaporation, as **flash type,** in which the vapor " flashes " from a body of hot water injected into a chamber under a partial vacuum; **film type,** in which a thin film of water passes over the surface of tubes filled with steam, and **submerged tube,** in which heat is imparted to the raw water by conduction from submerged tubes filled with steam.

If evaporation takes place in one stage the apparatus is designated as having a **single-effect.** A **double-effect** distilling plant is one in which the vapor from one evaporator, called the **first effect,** is condensed in the heating coil of another, called the **second effect,** the vapor from the latter being condensed in the usual way. In a **triple-effect** distilling plant the same process takes place in three stages; in a **quadruple-effect** plant there are four stages, and so on up to as high as twelve stages. In order that the heat may pass through the evaporator heating surface into the raw water, it is necessary to maintain a steam temperature higher than that of the water. Within the usual working limits the heat transferred will increase directly with the temperature difference so that the greater the temperature difference the less heating surface will be required to produce a given amount of vapor. It will, therefore, be seen that a multiple-effect evaporating plant will produce no more distilled water than a single-effect evaporator of the same size as one of the multiple-effect units, provided initial steam pressures and final vapor pressures are the same in both cases; but the pounds of distillate obtained per lb. of steam will be increased as the number of effects increases. While exact figures vary considerably with conditions, the distillate produced per lb. of steam averages about 0.8 lb., 1.5 lb., 2 lb. and 2.4 lb., for one, two, three and four effects, respectively. If the heat of the vapor were rejected to waste, the number of effects would be limited only by the investment costs, but the boiler feedwater can always absorb a certain amount of heat so the saving effected by the utilization of the heat rejected at the last stage must be taken into consideration. Ordinarily two effects are used, aside from any theoretical considerations, to provide needed flexibility and to permit operation when one effect is down for cleaning.

Figure 401 shows a section through the evaporating chamber of an Elliott evaporator illustrating the general principles of the flash type. Raw water is heated in an ordinary exhaust-steam closed feedwater heater and discharged into the chamber over a series of baffles. In flowing over these baffles a large water surface is exposed. Part of the water is evaporated because the pressure in the chamber is maintained below the boiling point and the rest falls to the bottom. The water at the bottom, plus

whatever makeup is necessary to supply the evaporation, is pumped through the heater and then back again to the evaporating chamber. The baffles remove whatever moisture may be entrained with the vapor. The latter is discharged into the main condenser or to a special **distiller condenser** so that a vacuum is always maintained in the chamber. Discharging the vapor into the main condenser unit is very uneconomical from purely a heat standpoint because the heat content of the vapor above that of the main unit condensate is rejected to the circulating water. The impurities in the raw water are blown off when a certain degree of concentration is reached. The amount of water evaporated is approximately 1 per cent of that circulated for each ten degrees temperature depression.

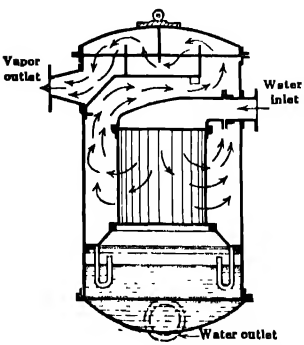


FIG. 401. Single-effect Flash Evaporator (Elliott).

Example 66. — Raw water at a temperature of 208 deg. fahr. is pumped into the chamber of a flash-type evaporator in which a vacuum of 24 in. (referred to 30-in. barometer) is maintained. What percentage of the water circulated is flashed into vapor.

Solution. — Temperature corresponding to 24-in. vacuum is 141 deg. fahr. Heat given up by each lb. of water due to temperature drop is $208 - 141 = 67$ B.t.u. Latent heat of steam at 141 deg. fahr. is 1012.6 B.t.u. per lb. Therefore, the weight of water evaporated per lb. of circulating water will be $67 \div 1012.6 = 0.066$, or 6.6 per cent, corresponding to approximately one per cent for each 10 deg. temperature drop. Since the latent heat does not vary much with the temperature or pressures at pressures below atmospheric the same temperature depression would produce practically the same percentage of flash irrespective of the actual initial temperature.

In the majority of evaporating plants the heating of the raw water is carried on in the evaporating chamber and not in a separate heater. Figure 402 shows a section through a **Lillie evaporator** illustrating the film type of construction. The evaporating chamber is partially filled with a nest of steam tubes which constitute the heating elements. Raw water, the level of which is maintained below the lowest row of tubes so that they are at no time submerged, is taken up by a centrifugal pump from the " pump well," discharged into the top of the shell and distributed over a perforated spray plate. From this plate it falls down over the tubes in a thin film, part of it flashes into vapor and the rest drops to the bottom of the chamber. The water in the pump well plus the required makeup is circulated over and over again. Condensate from the steam tubes is pure distilled water (provided the steam used for heating is free from impurities), and may be used as part of the feed supply.

Figure 403 shows a section through a **Reilly evaporator** illustrating the

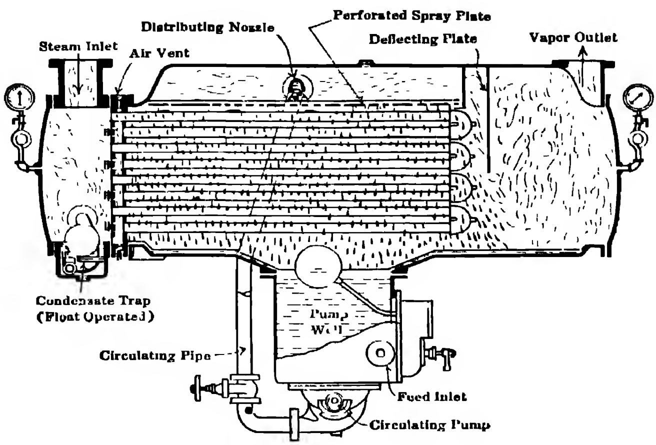


FIG. 402. Lillie Film-type Evaporator.

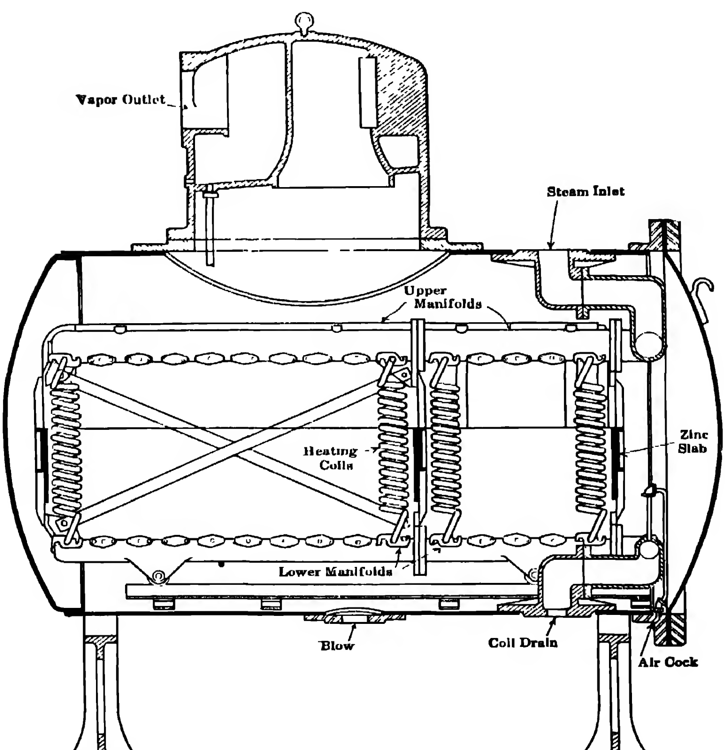


FIG. 403. Submerged-tube Type Evaporator. (Reilly.)

submerged-tube type. The heating element is composed of a series of helical coils connected to suitable manifold headers and completely submerged in the raw water. Steam enters the vapor manifold, passes downward through the coils to the lower manifolds, condenses, and is discharged as condensate from the coil drain. Raw water is fed into the chamber near the bottom of the shell at such a rate as to maintain a constant level. The vapor boiled off passes to the condenser in the usual manner.

Example 67. — Calculate the amount of raw water evaporated per lb. of steam supplied in a (1) single-effect; (2) double-effect and (3) triple-effect evaporator of the submerged-tube type if the conditions are as follows: Pressure of supply steam, 17 lb. abs., vacuum in last effect 25.92 in. (referred to 29.92 barometer); temp. of raw water, 50 deg. fahr. Neglect all losses.

Solution. — (*Single-effect*, Fig. 404.)
From steam tables, we find

	Steam Supplied	*Vapor Discharged*
Pressure, lb per sq. in. abs.	17	4 (in. of mercury)
Corresponding temperature, deg. fahr.	219.4	125.5
Latent heat, B.t.u. per lb.	965.6	1021.3

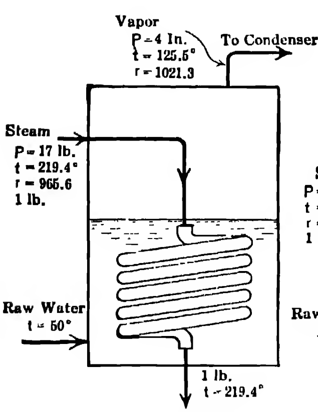


Fig. 404.

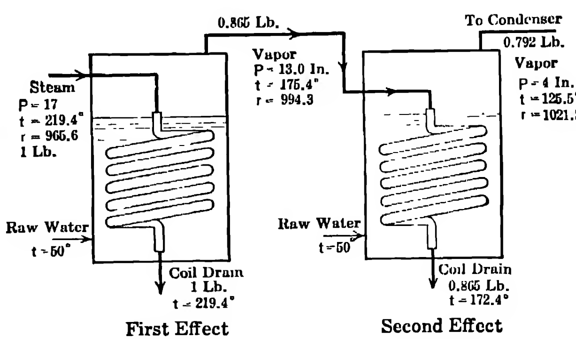


Fig. 405.

Heat given up by 1 lb. of steam condensed in coils = 965.6 B.t.u.
Heat absorbed by 1 lb. of vapor leaving evaporator = 1021.3 + (125.5 − 50) = 1096.8 B.t.u.
Weight of raw water vaporized per lb. of steam supplied = 965.6 ÷ 1096.8 = 0.884 lb.

Double-effect, Fig. 405. Assume the temperature drop between steam supply and the condenser to be equally divided between stages. Total drop = 219.4 − 125.5 = 93.9 deg.; drop in each effect = 93.9 ÷ 2 = 46.95 deg. Therefore, temperature of first effect = 219.4 − 46.95 = 172.45

deg., and that of second effect = 172.45 − 46.95 = 125.5. From steam tables:

	Steam Supplied	*First Effect*	*Second Effect*
Pressure, in. abs.	17 (lb. per sq. in.)	13.01	4
Temperature, deg. fahr.	219.4	172.45	125.5
Latent heat, B.t.u. per lb.	965.6	994.3	1021.3

Heat given up by 1 lb. of steam entering first stage = 965.6 B.t.u.
Heat absorbed by vapor discharged into second stage = 994.3 + (172.45 − 50) = 1116.7 B.t.u.
Water evaporated from first effect = 965.6 ÷ 1116.7 = 0.865 lb.
Heat given up by vapor entering coils of second effect = 994.3 B.t.u. per lb.
Heat absorbed by vapor discharged into second stage = 1021.3 + (125.5 − 50) = 1096.8.

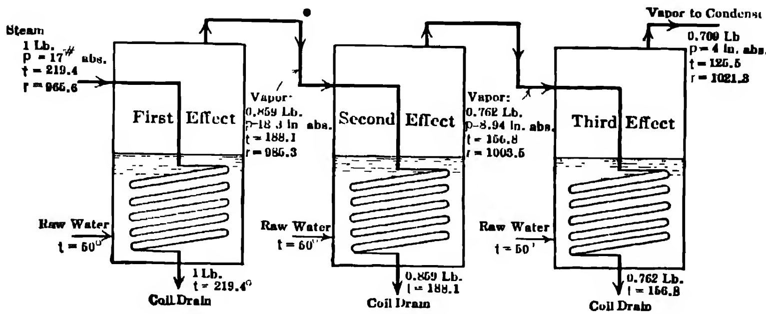


Fig. 406.

Water evaporated from second effect per lb. of vapor supplied to the coils = 994.3 ÷ 1096.8 = 0.916 lb.
Water evaporated from second effect per lb. of steam = 0.865 × 0.916 = 0.792 lb. Total evaporation per lb. of steam = 0.865 + 0.792 = 1.657 lb.

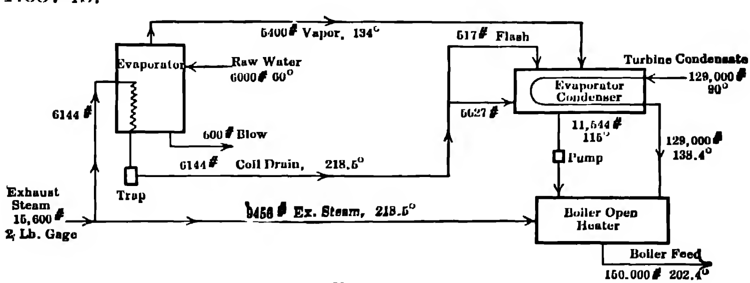


Fig. 407.

Triple Effect, Fig. 406. Proceed as for double effect, assuming that the temperature drop in each stage will be one-third of the total. The various calculated quantities are given in the illustration.

Figure 407 shows the heat exchange in a single-effect evaporator system

various tube passages by an induced-draft fan or, occasionally, by stack draft. Feedwater is forced into the economizer through the lower branch pipe nearest the point of exit of gases, and emerges through the upper branch nearest the point at which the gases enter. Each tube is encircled with a set of triple overlapping scrapers which travel continuously up and down the tubes at a slow rate of speed, the object being to keep the external surfaces free from soot and fine ash deposit. The mechanism for working the scrapers is placed on top of the economizer, outside the chamber, and the power is supplied either by a belt from some convenient shaft or by a small independent engine or motor. The power for operating the gearing varies from 1/2 to 1 hp. per 1000 sq. ft. of economizer surface, depending upon the number and length of tubes. The apparatus is fitted with blow-off and safety valves, and a space is provided at the bottom of the chamber for collecting the soot. In some designs, the outer surface of the tubes is cleaned by steam jet blowers similar to soot blowers, and, in others water sprays are used for flushing away the deposits. Soot accumulations may be mechanically removed from the bottom of the chamber, as shown in the illustration, or by means of vacuum systems.

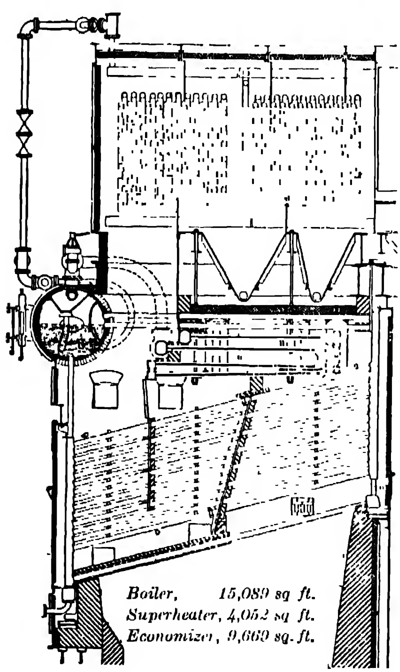


FIG. 409. B. & W. Transverse-tube Economizer, Calumet Station.

Figure 409 shows an assembly of a Babcock & Wilcox **transverse tube,** and Fig. 410 a similar view of a Babcock & Wilcox **transverse counterflow** economizer, illustrating the modern steel-tube type. The unit illustrated in Fig. 409 is composed of a number of B. & W. sections similar in design to those of the boiler. It is placed above the boiler and arranged with the tubes transversely to the boiler tubes, thus giving a single pass over the economizer surface. A number of sections are placed side by side, the feedwater being forced through each in series, thereby approximating counterflow. This design may be constructed with either 4-in. or 2-in. tubes. The economizer shown in Fig. 410 differs from the one just described in that forged-steel square boxes are used instead of the standard

B. & W. sections. A single row of tubes from the bottom box on one side of the economizer passes to a corresponding box on the other side. From the latter a second row of tubes passes to the second box on the feed inlet

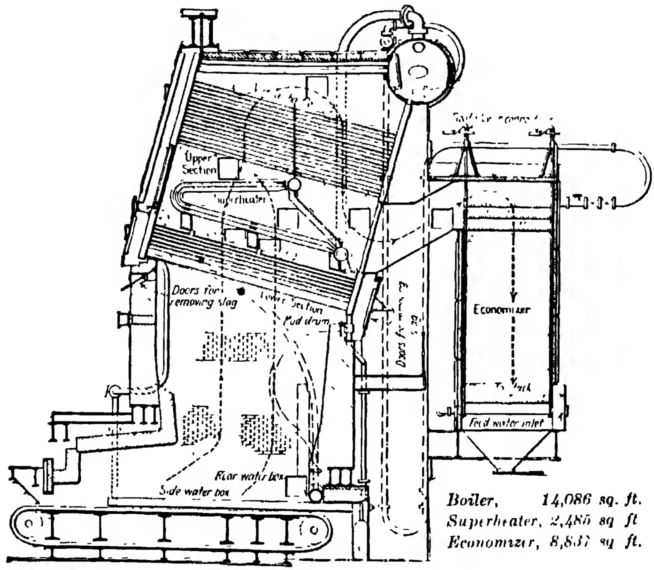


FIG. 410. B. & W. Transverse Counterflow Economizer, Waukegan Station.

side, and in this manner the water passes throughout the whole economizer from bottom to top. The gases flow from top to bottom, giving a true counterflow. Placing the economizer at the back of the boiler offers a very compact arrangement of boiler and economizer and allows the induced-draft fan apparatus to be located either on the boiler room floor or in the basement. Steel-tube economizers are usually equipped with steam jet blowers for soot removal.

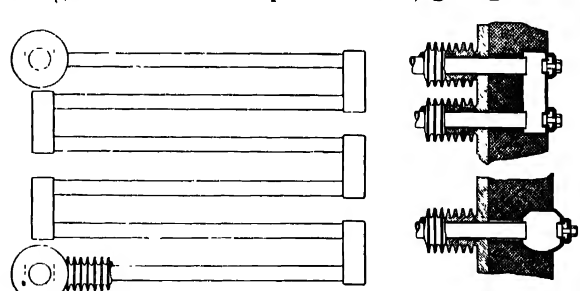


FIG. 411. Foster Economizer Element.

Figure 411 gives the general details of a **Foster** economizer, which is in a measure similar to the Foster superheater. The heating surface is composed of a number of 2-in. steel tubes expanded into cast-steel return

headers, and fitted on the outside with a series of cast-iron gilled rings. The lower and upper banks of tubes are expanded into forged steel manifolds, as indicated. The elements are always placed horizontally, the gases passing directly across the tubes and the water passing in series through each bank, thus giving a true counter-current effect. The steel

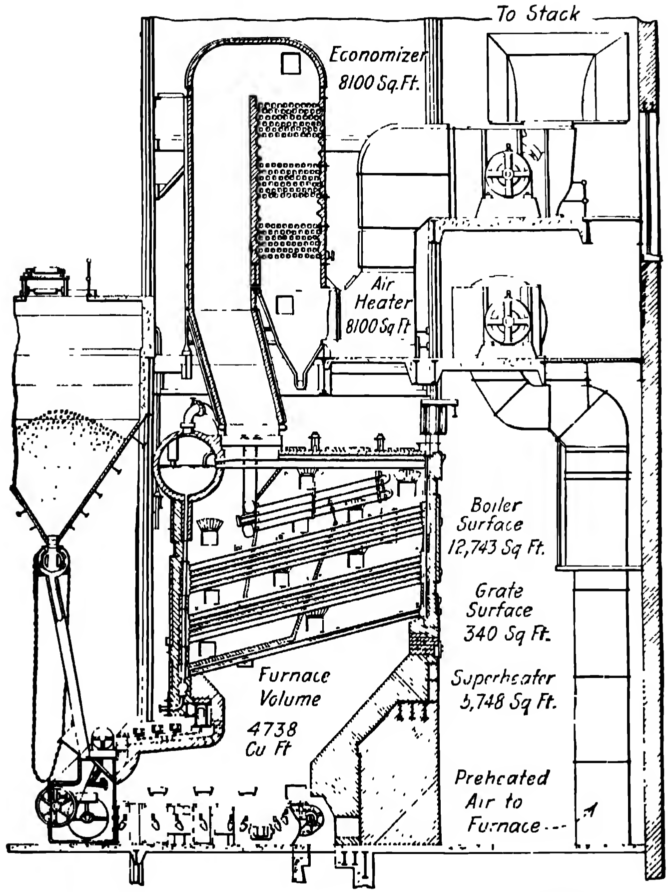


FIG. 412. Foster Economizer and Air Heater Installation, Northeast Station.

tubes permit of high pressures, and the cast-iron envelopes insure resistance against external corrosion. The extended surface also increases the heat transmission. The points between the cast-iron rings are so tight that no water can reach the outer surface of the steel tubes when soot is removed by washing, which is the method adopted for cleaning.

Figure 412 shows a diagrammatic arrangement of the Heine "Type

S " boiler equipment at the Northeast Station of the Kansas City Power and Light Co. This is one of the first large central stations to adopt an air heater in conjunction with an economizer.

Figure 413 shows a section through one of the 25,000 sq. ft. Badenhausen boilers, as installed in the Highland Park Plant of the Ford Motor Co., illustrating an economizer element integral with the boiler. Feedwater enters drum 6, flows down the rear bank and enters the forward bank of tubes connecting drums 5 and 6. The economizer element is baffled so that the gases are forced to travel down the front and up the rear bank of tubes. The resulting difference in temperature creates a positive circulation of the water in the economizer element. The advantages claimed for the boiler with integral type of economizer or preheater boiler are: (1) it is in the same setting as the boiler; (2) it requires no by-passed breeching; (3) it necessitates no additional overhead structure or rear space; (4) it has at least 5 per cent higher efficiency than any boiler without the preheater; and (5) the first cost of a properly proportioned boiler with preheater is no greater than that of a standard boiler of equivalent capacity.

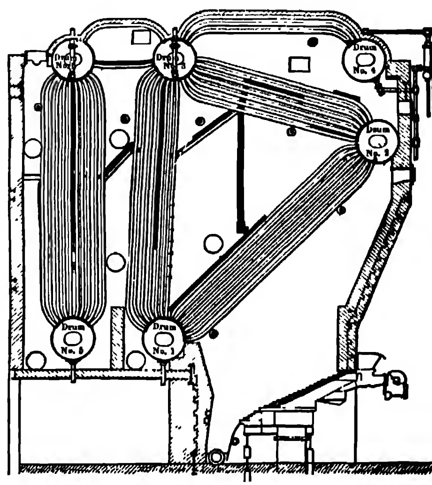


FIG. 413. Badenhausen Boiler with Preheater Element.

262. Temperature Rise in Economizers. — The heat transfer in an economizer follows the same basic law as the heat transmission through any heating surface, viz:

$$SUd = w_1c_1\ (t - t_0), \tag{245}$$

$$= w_2c_2\ (t_2 - t_1), \tag{246}$$

in which

S = total heating surface, sq. ft.,

U = mean coefficient of heat transmission, B.t.u. per hr. per sq. ft. per deg. mean temperature difference,

d = mean temperature difference between the two fluids, deg. fahr.,

w_1 and w_2 = weights, respectively, of the fluid to be heated and the flue gas,

c_1 and c_2 = mean specific heats respectively of the fluid to be heated and the flue gas,

t_0 and t = initial and final temperature of the fluid to be heated, deg. fahr.,

t_2 and t_1 = initial and final temperature of the flue gas, deg. fahr.

By an analysis similar to that developed in paragraph 242, it may be shown that for either parallel flow or counterflow

$$d = (t_i - t_f) \div \log_e (t_i/t_f) \tag{247}$$

in which

t_i, t_f = initial and final temperature difference between the two fluids.

Arithmetic mean temperature difference, $d_m = (t_1 + t_2)/2 - (t + t_0)/2$.

By combining equations (245) to (247) and reducing (see Sibley Journal, Jan., 1916, p. 129), we have as an expression for the temperature rise in the fluid for parallel flow

$$x = (t_2 - t_0) \div \left(\frac{N - 1}{10^n - 1} + N\right) \tag{248}$$

in which

x = temperature rise in the fluid, deg. fahr.,
$N = w_1c_1/w_2c_2$
$n = SU\ (N - 1)/2.3w_1$

Other notations as previously designated.

Example 68. — Calculate the final feedwater and flue-gas temperature for a cast-iron economizer installation operating under the following conditions: Boiler heating surface 12,000 sq. ft.; economizer surface 7500 sq. ft.; initial feedwater temperature 100 deg. fahr. and initial flue-gas temperature 650 deg. fahr. when the boiler is operating at 100 per cent above standard rating; coal used, Illinois screenings, 11,400 B.t.u. per lb.

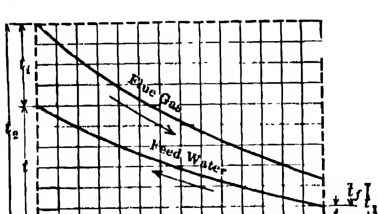


FIG. 414. Counter-Current Flow.

Solution. — It has been shown (paragraph 44) that the theoretical weight of air per lb. of any coal is approximately 7.5 lb. per 10,000 B.t.u. Therefore for the coal specified,

1.14 × 7.5 = 8.55 lb. = theoretical air requirements per lb. of coal.

Assuming an air excess of 50 per cent at maximum load and allowing 15 per cent for ash, the probable actual weight of flue gas per lb. of coal = 1.5 × 8.55 + 0.85 = 13.7 lb., or in round numbers 14 lb.

Since the evaporation at rating is equivalent to 3.45 lb. from and at

212 deg. per sq. ft. heating surface per hr., at 100 per cent overload the total weight of water, w, fed to the boiler is

$$w = 2 \times 12{,}000 \times 3.45 = 82{,}800 \text{ lb. per hr.}$$

Assuming an overall efficiency of 75 per cent, the weight of coal required is

$$\frac{970.4 \times 82{,}800}{11{,}400 \times 0.75} = 9400 \text{ lb. per hr.}$$

The total weight of flue gas, w_2, is

$$w_2 = 9400 \times 14 = 131{,}600 \text{ lb. per hr.}$$

Assume the mean specific heat of the water to be unity and that of the flue gas to be 0.25.

Assume $U = 4.25$, which is an average value for a cast-iron economizer with initial flue-gas temperature of 650 deg. fahr. Substituting these values in equation (248)

$$N = \frac{w_1c_1}{w_2c_2} = \frac{82{,}800 \times 1}{131{,}600 \times 0.25} = 2.52.$$

$$n = \frac{SU(N-1)}{2.3w_1} = \frac{7500 \times 4.25\ (2.52-1)}{2.3 \times 82{,}800} = 0.254.$$

$$x = \frac{t_2 - t_0}{\dfrac{N-1}{10^n - 1} + N} = \frac{650 - 100}{\dfrac{2.52-1}{10^{0.254} - 1} + 2.52} = 126 \text{ deg. fahr.}$$

Since $x = t - t_0$, the final temperature of the feedwater is

$$t = 126 + 100 = 226 \text{ deg. fahr.}$$

The heat absorbed by the feedwater must be equal to that given up by the flue gas, or

$$w_1c_1\ (t - t_0) = w_2c_2\ (t_2 - t), \tag{249}$$

from which

$$(t_2 - t_1)/(t - t_0) = w_1c_1/w_2c_2 = N. \tag{250}$$

Substituting the known quantities in equation (250)

$$(650 - t_1)/(226 - 100) = 2.52$$

or

$t_1 = 332.5$ deg. fahr. = final temperature of the flue gas.

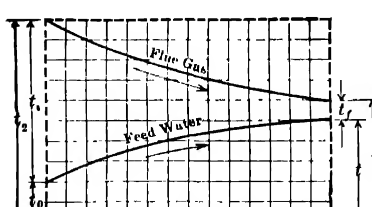


FIG. 415. Parallel-Current Flo[w]

For parallel flow, as in Fig. 415, the final flue-gas temperature may be calculated from the following formula which was deduced from equations (245) to (247).

$$t_1 = (t_2 - a/b) \div e^m + a/b \tag{251}$$

in which

$$a = w_1c_1t_2/w_2c_2 + t_0$$
$$b = w_1c_1/w_2c_2 + 1$$
$$m = bSU/w_2c_2$$

Other notations as previously designated.

In view of the uncertainty in practice of many of the factors entering into the preceding analysis, it is sufficiently accurate for most purposes to take the algebraic mean value of d instead of the logarithmic value. This greatly simplifies the calculations. In using the algebraic mean temperature difference, the *corresponding* value of U must be taken and not that based on logarithmic temperature difference.

263. Factors Determining Installation of Economizers. — The general conclusion drawn from current practice is that an economizer installation results in:

(1) A saving in fuel ranging from 7 to 20 per cent.

(2) Overall boiler, superheater and economizer efficiencies ranging from 80 to 93%. (91.6% overall efficiency has been obtained at the Lakeside plant and 93% is expected at the new Richmond Station.)

(3) A substantial overall gain in heat economy where the boilers are operated with high flue-gas temperatures.

(4) Maximum overall heat economy when the boilers are forced far above their rating and the auxiliaries are electrically driven, and pure feedwater is available. This is true only for the conventional type of turbine or engine. If the feedwater is heated by bleeding the prime mover, the value of the economizer becomes less as the temperature of the feedwater is increased.

(5) Decreased wear and tear on the boilers due to the high feedwater temperature.

(6) A large storage of hot water for sudden peak demands.

Against these advantages may be cited:

(1) Increased first cost and maintenance of economizer elements.

(2) Increased power requirements of induced-draft fan.

(3) Treatment of feedwater to prevent internal corrosion.

(4) Extra space requirements for economizer equipment.

In the large modern central station, where the steam pressures are very high and boilers are operated continuously at heavy ratings and the fuel is costly, feedwater economizers have definitely proven their worth, except, perhaps, where the feedwater is heated to a high temperature by stage bleeding. The tendency in the latter situation is to reclaim the waste heat from the flue gases by means of air preheaters.

Some of the more important factors to be considered before installing a feedwater economizer are:

(1) *Cost of Fuel.* — The higher the cost of fuel the greater will be the returns. By equating the thermal gain against first cost and maintenance at various loads, the net economic gain may be estimated. In the small plant with variable loads, the net thermal gain is usually offset by the additional fixed charges and maintenance costs.

(2) *Temperature of the Flue Gas.* — Since the primary object of an economizer is the utilization of the heat rejected to the chimney, it stands to reason that the value of an economizer installation increases with the temperature of the flue gases. In the modern high-pressure boiler for central stations, attempt is no longer made to reduce the gases to the lowest point possible with little regard for the amount of boiler surface required to effect this result. On the contrary, the present idea is to design the boiler so that all of the surface will be subjected to a much higher temperature difference and thereby increase the rate of heat transfer. This necessitates the gases leaving the boiler at from 50 to 100 deg.

TABLE 84

TESTS OF FOSTER ECONOMIZERS

A. Delray Station, Detroit Edison Co.

Heating surface of Class W53 Stirling Boiler	23,654 sq. ft.
Heating surface of Class W57 Stirling Boiler	12,054 sq. ft.
Heating surface of B. & W. Superheater, large boiler	2,217 sq. ft.
Heating surface of B & W. Superheater, small boiler	1,528 sq. ft.
Heating surface of Foster Economizer	18,800 sq. ft.
Ratio of economizer heating surface to that of boilers	52 6 per cent

Date	1922	Apr 20	Apr 18	Apr 19	Apr. 10	Apr. 17
Duration	hr.	2	3	3	3	3
Power developed by unit	boiler hp.	3859	5780	5867	6955	6978
Percentage of rated capacity	per cent	108	162	164	194	196
Steam pressure by gage	lb per sq. in	210	212	212	213	215
Superheat	deg. fahr.	147	171	156	169	185
Gas temperature at economizer inlet	deg fahr.	506	563	566	566	631
Gas temperature at economizer outlet	deg. fahr.	276	313	301	296	337
Gas temperature drop through economizer	deg. fahr.	230	250	265	270	294
Water temperature at economizer inlet	deg fahr.	205	201	206	193	194
Water temperature at economizer outlet	deg fahr.	272	273	283	282	296
Water temperature rise in economizer	deg. fahr.	67	72	77	89	102
Draft in flue leaving economizer	in. w. g.	2 18	3 66	4.16	5 35	5.31
Draft in flue entering economizer	in. w. g.	60	67	82	.97	.81
Draft loss through economizer	in. w g.	1 58	2.99	3.34	4.38	4 50
Fuel saving effected by economizer	per cent	6.03	6 39	6.90	7.85	8.93

B. Riverside Station, Municipal Gas Co., Albany, N. Y.

Heating surface of special water-tube boiler	6877 sq. ft
Heating surface of Foster Radiant Heat Superheater	65 2 sq. ft.
Heating surface of Foster Economizer	5856 sq. ft.
Grate surface of stoker	105 sq. ft.
Ratio of economizer heating surface to that of boiler	85.1 per cent

Date	1921	July 27	Aug. 9	July 28	July 2
Duration	hr.	6	6	6	69
Power developed by unit	boiler hp.	805	935	1016	1201
Percentage of rated capacity	per cent	117	136	148	175
Steam pressure by gage	lb per sq. in.	200	202	201	202
Superheat	deg. fahr	188	129	114	137
Gas temperature at economizer inlet	deg. fahr.	582	663	646	697
Gas temperature at economizer outlet	deg fahr	248	270	250	267
Gas temperature drop through economizer	deg. fahr.	334	393	396	430
Water temperature at economizer inlet	deg fahr.	213	213	212	212
Water temperature at economizer outlet	deg. fahr.	334	356	354	358
Water temperature rise in economizer	deg fahr.	121	143	142	146
Draft in flue leaving economizer	in. w. g.	50	1 20	1 20	1.50
Draft in flue entering economizer	in. w. g.	+ 05	− 07	+ 01	− 19
Draft loss through economizer	in. w g	55	1 13	1 21	1 31
Fuel saving effected by economizer	per cent	10 78	13 08	13 09	13 30

higher than formerly. The gases are then reduced to 200 to 300 deg. by the economizer. The amount of economizer surface ranges from 60 to 100 per cent of the boiler heating surface, depending upon the method of establishing the heat balance.

(3) *Temperature of the Feedwater.* — The water entering the economizer should have a temperature at least equal to that corresponding to the dew point of the gases, to prevent external corrosion, and in case the water is not deaërated the temperature should not be less than 180 deg., to prevent internal corrosion. With properly degasified water, inlet temperatures as low as 120 deg. fahr. have worked out satisfactorily, but the exact temperature is dictated by the station heat balance.

(4) *Pressure Drop through Economizer.* — For high rate of heat transference, the gases should flow through the economizer at high velocities; but, as the pressure drop increases approximately as the square of the velocity, the gain in heat transfer is at the expense of increased power requirements for the fan. The pressure drop varies widely with the design of economizer and the temperature of the gases, but an average value is 0.5 in. of water at 20 ft. per sec. to 3 in. at 50 ft. per sec. The mean

coefficient of heat transfer varies with the weight of gas flowing per hr., the mean temperature difference between the gas and that of the water, the composition of the gas, cleanliness of the tubes, and design of economizer, and ranges from 2 to 7 B.t.u. per hr. per sq. ft. per deg. fahr. difference in temperature. The curves in Figs. 416 and 417 give the results of a series of tests conducted on the 4800 sq. ft. Foster steel-tube economizer installed in the 6865 sq. ft. Stirling boiler at the Seventieth St. plant of the Cleveland Elec. Ill. Co. and give some idea of the various factors involved in the performance of the economizer.

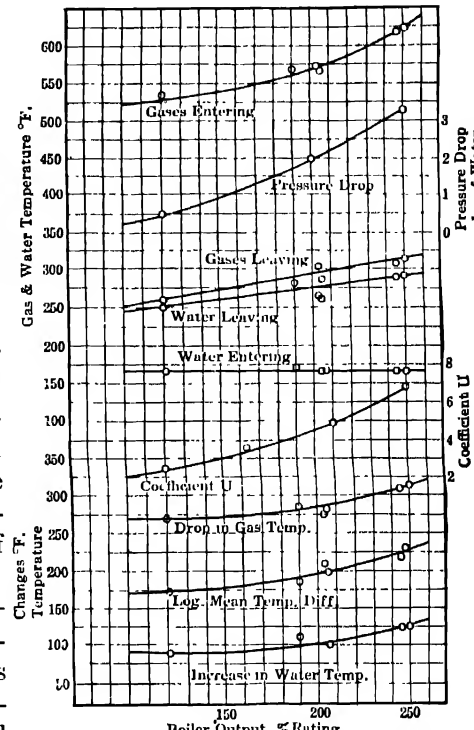


FIG. 416. Performance of Foster Steel-tube Economizer.

(5) *Purity of the Feedwater.* — With hard feedwater, the formation of scale within the tubes may seriously affect the efficiency of heat transmission, and the cost of cleaning may prove excessive. Internal corrosion may also be caused by dissolved gases in the feedwater.

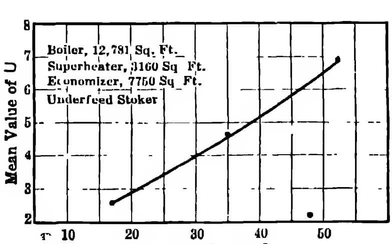


FIG. 417. Influence of Gas Velocity on Coefficient of Heat Transfer.

(6) *Minimum Temperature of the Flue Gas.* — The flue-gas temperature should not be lowered below the dew point, since the condensation of the vapor content may cause the soot to adhere to the tubes and render its removal a costly problem. An average minimum is 240 deg. fahr. With coals high in sulphur content, the moisture forms sulphuric acid which corrodes the tubes.

(7) *Increased Capacity Due to the Additional Heating Surface.*

(8) *Cost of Additional Building Space.*

(9) *Cost of Producing the Draft.* — For chimney draft, this means cost

of the extra height of stack necessary to overcome the loss in draft. This may range from 20 to 40 per cent of the total cost of the chimney. In the modern mechanical draft installation, the power required to operate the fans ranges from 1 per cent to 4 per cent of the main generator output.

(10) *First Cost.* — See Chapter XIX.

(11) *Boiler Pressure.* — Cast-iron superheaters are used for working pressures as high as 400 lb. per sq. in., but the cost increases rapidly with increase in pressure above 250 lb. per sq. in. All modern superheaters for pressures varying from 300 to 1200 lb. per sq. in. are of steel-tube construction.

264. Flue-gas Combustion-air Heaters. — Many of the latest power plant installations include combustion air heaters, or **air preheaters.** Their value for increasing power plant efficiency has been demonstrated and they have come to take an important part in the steam power station. There are three types of air preheaters in American practice: (1) plate, (2) tubular, and (3) regenerative. The **plate-type** heater consists of a steel casing in which are contained a number of sheet-iron envelopes as shown in Fig. 418, the flue gas and air passing through alternate envelopes in opposite directions. In the **tubular** type the air is passed countercurrently over the outside of tubular elements and the flue gas is exhausted through them. The **regenerative** type consists of a rotating element which alternately absorbs heat from the flue gas and transfers it to the air. The Combustion Engineering Corporation heater is a well-known example of the plate type, the Babcock & Wilcox heater of the tubular type, and the Ljungstrom of the regenerative type. The mean coefficient of heat transfer in all types of air preheater is low, ranging from 1 to 4 B.t.u. per sq. ft. per deg. fahr. temperature difference.

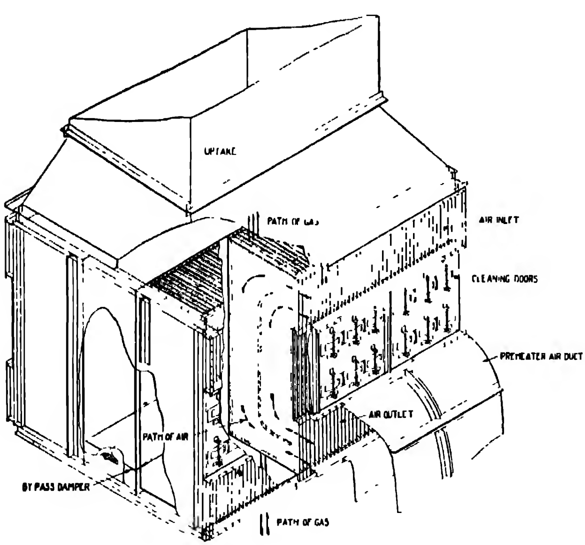


FIG. 418. Air Preheater Combustion Engineering Corporation.

The curves in Fig. 58 give the performance of Boiler No. 9 at the Colfax Station of the Duquesne Light Co., as presented by C. W. E. Clarke, Power Engineer of the Dwight P. Robinson Co., before the Dec., 1923, meeting of the A.S.M.E., and serve to illustrate the results obtained for a specific set of conditions. The present indications are that air preheaters will be used extensively where the feedwater is raised to a high temperature by bleeding the main turbine. See also paragraph 50.

Present Status of Air Preheaters. Power, March 31, 1925, p. 486.
Air Preheaters and Their Application. Power, Dec. 2. 1924, p. 884.
Types of Air Preheaters. Power Plant Engrg., Sept. 15, 1925, p. 940.

265. Heat Balance. — The heat balance in steam power plants has been defined as "that correlation of events in the heat cycle of the plant as a whole which adjusts the rate of steam generation to the demand with the least possible waste of heat and at the least possible expenditure of fuel." Obviously, everything that has to do with the generation or absorption of heat is a factor in the heat balance, but in a general sense the principal factor is the means employed for heating the feedwater. There are so many possible combinations of prime movers, auxiliaries, and heaters in establishing a heat balance that it is futile to attempt to cover the subject in a book of this general nature and only a few of the more commonly used systems will be briefly described. Among the latter may be mentioned:

1. *Steam-driven auxiliaries used entirely, exhausting into open feedwater heaters.* — This system of feedwater heating was practically universal in all steam plants of a decade or more ago. For small plants with low load factors where there is no excess of exhaust steam over and above that required to heat the feedwater to 212 deg. fahr., this is perhaps the simplest and cheapest system. The two main objections to this system in ordinary practice are: (1) it is difficult, if not impossible, to maintain a heat balance under all load conditions; and (2) the heat energy of the steam available for work can be only partly utilized in small plants. It has been shown in paragraph 181 that, when exhaust steam is used for heating the feedwater, a kw-hr. is produced by that steam before exhaust at an expenditure of approximately 4000 B.t.u., or about one-third as much as with a highly efficient turbine-driven generator exhausting against a back pressure of 1 in. of mercury. It is obvious that the production of this cheap power is limited in amount by the ability of the feedwater to absorb the exhaust steam produced.

2. *Mixed steam and electrically-driven auxiliaries.* — In the majority of

[1] Mech. Engrg., Feb., 1924, p. 64.

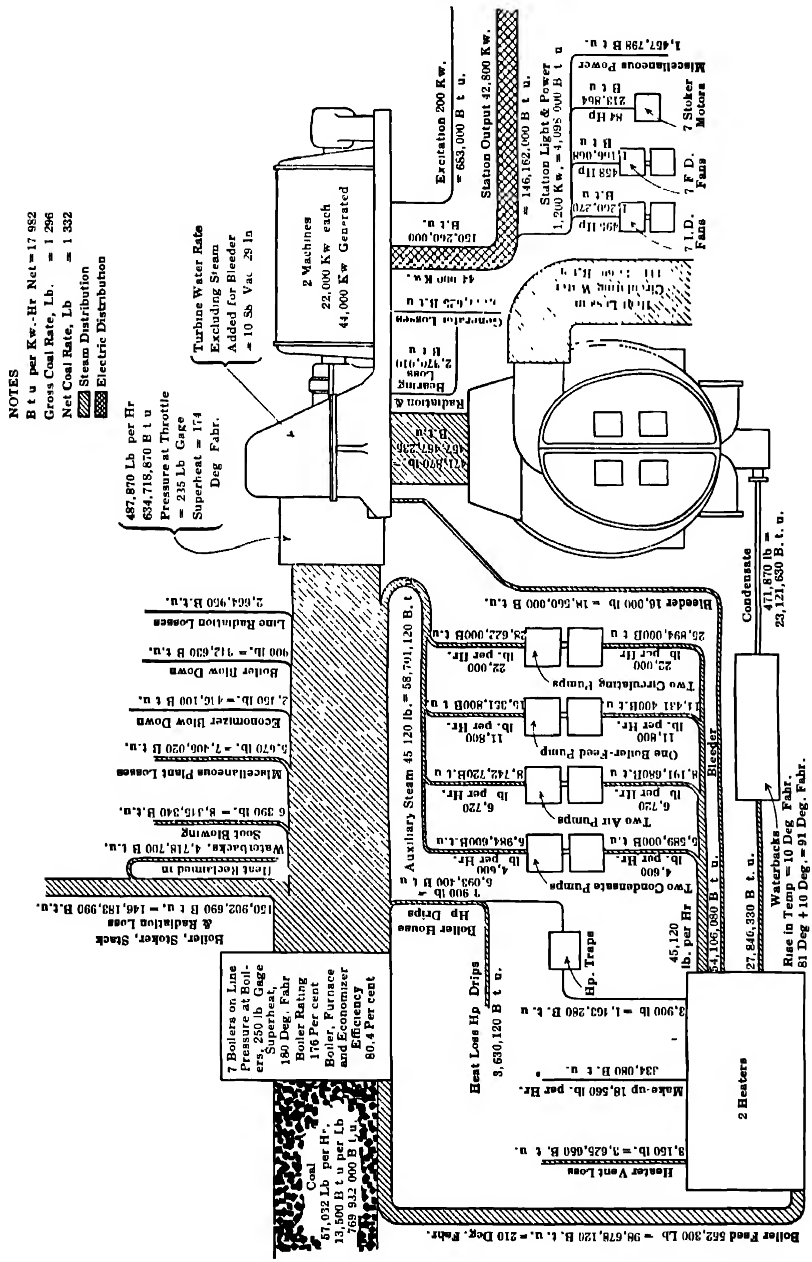


FIG. 419. Heat Balance at Most Economical Load. Delaware Station.

medium-sized plants and in a number of our modern central stations, part of the auxiliaries are motor-driven and part steam-driven, the motors taking current from the main generator. The division is such that a proper selection of each type of drive can be made at different loads to obtain a satisfactory heat balance.

For a detailed analysis of the heat balance for a large station of this class, consult "Auxiliary System and Heat Balance at the Delaware Station of the Philadelphia Electric Co.," *Trans. A.S.M.E.*, Vol. 43, 1921, p. 475. Figure 419 gives the heat balance of this station at the most economical load.

3. *The house, or auxiliary, turbine system.* — In this system the feedwater is heated by the exhaust from a small turbo-generator designated as the house turbine. The current generated is used to operate all or part of the electrically-driven auxiliaries and for other house service.

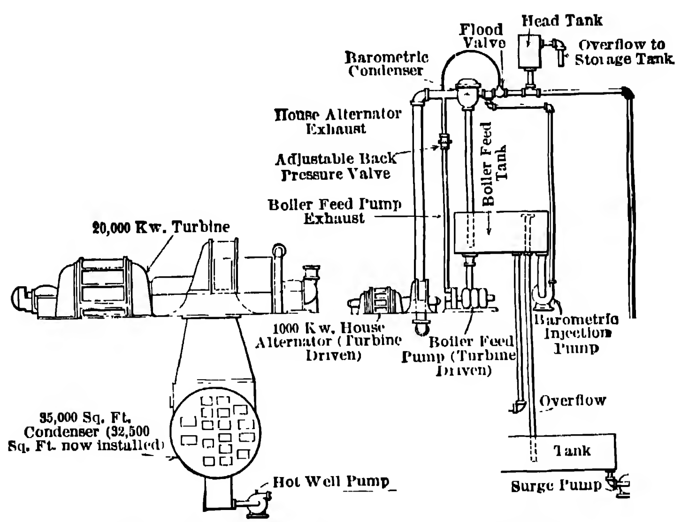


FIG. 420. House Turbine at Connors Creek Station.

The turbine exhausts into an open condenser, the circulating water for which is the feedwater. This condenser is, therefore, the equivalent of an open heater in which the exhaust from the house turbine (and other steam-driven auxiliaries, if any) mixes with and heats the condensate from the main units. This system is used in a number of large central stations. It has the advantage, over the other systems, of providing a simple method of degasification without the addition of deaërating or deactivating equipment. House turbines are installed

(*a*) in conjunction with other steam-driven auxiliaries or a combination of steam and electrically-driven auxiliaries. The Connors Creek Station

of the Detroit Edison Co. is an example of this system, a diagrammatic arrangement of which is illustrated in Fig. 420. For a detailed analysis of the heat balance in this station, consult *Trans. A.S.M.E.*, Vol. 43, 1921, p. 500. Consult also, " Heat Balance at Colfax," *Trans. A.S.M.E.*, Vol. 43, 1921, p. 487.

(*b*) in conjunction with electrically-driven auxiliaries operated with current from the house turbine or the main unit, or with the supply divided between the house turbine and main unit. This system is adopted at the Hell Gate Station of the United Electric Light and Power Co. For an analysis of the heat balance of this plant, consult *Trans. A.S.M.E.*, Vol. 43, p. 495.

(*c*) same as (*b*) except that deficiency in house turbine exhaust is made up by bleeding from the lower stages of the main turbine units. This system is adopted at the Hudson Ave. Station of the Brooklyn Edison Co.

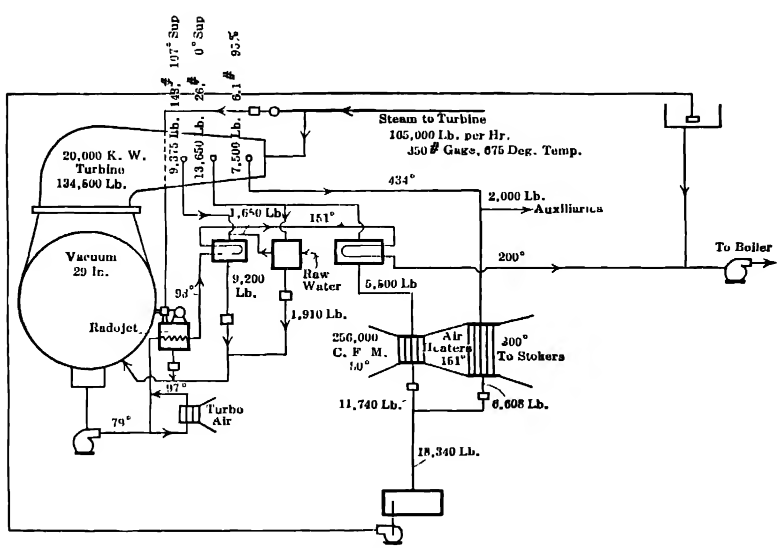


FIG. 421. Heat-flow Diagram. Air Preheater Installation.

4. *Auxiliaries all electrically driven and the feedwater heated by bleeding steam from intermediate stages of the main turbine.* — This is now accepted as the most efficient method for large central stations. The main unit is the most efficient user of steam; therefore, the most convenient method of obtaining the maximum electrical output for a given amount of exhaust steam is by bleeding it at one or more stages. The advantage of bleeding in stages instead of all at one point follows from the fact that the steam used for heating the condensate at the lowest temperature can do more

work in the turbine than if all steam were bled at a higher point. Bleeding steam from intermediate stages also has the advantage of relieving congestion in the lower stages of the machine. Some idea of the economy of bleeding the main turbine may be gained from the following approximate values: 1 kw-hr. will be produced for every 14 lb. bled at 150 deg. fahr. from a large modern unit as against 20 lb. with a house turbine

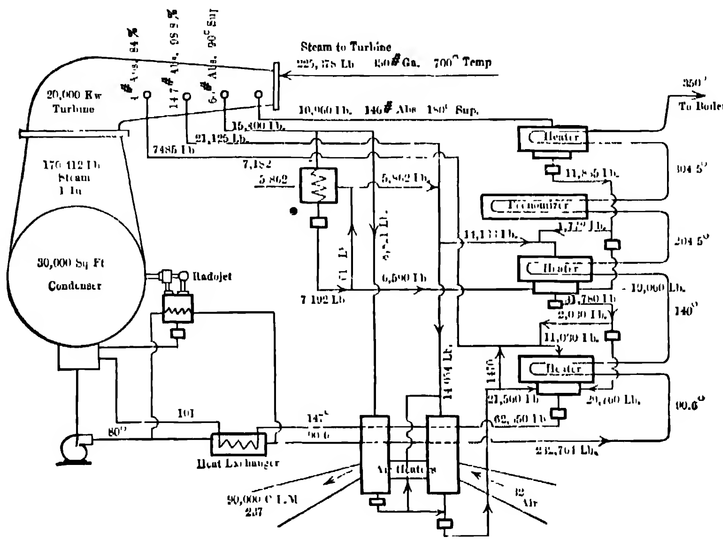


FIG. 422 Heat-flow Diagram. Air Preheater and Economizer Installation.

exhausting at the same temperature; 1 kw-hr. will be produced for every 17 lb. bled from the main unit at 210 deg. fahr. as against 24 lb. with a house turbine, and 40 lb. with the ordinary type of good auxiliary turbine exhausting at the same temperature. The water rate, neglecting leakage and heat losses, for any other temperature may be approximated by dividing 3415 by the heat equivalent of the theoretical work done in expanding adiabatically from the given initial conditions to the temperature of the feedwater and dividing the result by the Rankine-cycle ratio, thus: For 340 lb. abs., 200 deg. superheat, 120 deg. fahr. feedwater temperature, and Rankine-cycle ratio of 0.68, the theoretical work done per lb. is $1323 - 862 = 461$ B.t.u. and the actual water rate per kw-hr. $= 3415 \div (461 \times 0.68) = 10.9$ lb. Leakage and heat losses will probably bring this up to 11 or 11.2 lb.

For any set of conditions there exists a definite feedwater temperature at which the efficiency of power generation is a maximum. This is shown by the curves in Fig. 423, which give the theoretical values obtained from steam extracted at a single stage and subsequently used for heating

feedwater. This curve is based on 340 lb. abs. initial pressure with 200 deg. fahr. superheat and condensate temperature of 79 deg. fahr. The work done for, say, the point 300 deg. is calculated as follows: Heat content of 1 lb. of steam at 340 lb. abs. and 200 deg. superheat is 1323 B.t.u. Heat content after adiabatic expansion to 300 deg. fahr. is 1170. Work done by 1 lb. of steam = 1323 − 1170 = 153 B.t.u. Net heat

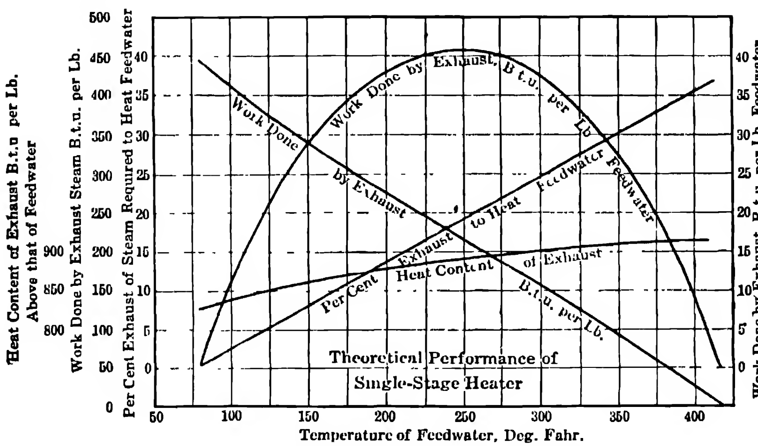


FIG. 423. Theoretical Performance of Single-stage Heater.

content of the exhaust above 300 deg. is 1170 − 270 = 900 B.t.u. per lb. Pounds of exhaust steam required to heat 1 lb. of feedwater from 79 to 300 deg. fahr. = (270 − 47) ÷ 900 = 0.248. Work done by 1 lb. of exhaust per lb. of feedwater = 0.248 × 153 = 37.6 B.t.u. From the curves it is apparent that the best results are obtained theoretically for a feedwater temperature of approximately 250 deg. fahr. with a condensation of about 19 per cent of the steam entering the turbine. These curves are based on a 100 per cent Rankine-cycle ratio and no leakage or heat losses. By considering the Rankine-cycle ratio at each back pressure and including the heat and leakage losses, the probable actual temperature curve may be readily drawn. Complications in piping and control, and the fact that no steam auxiliaries are available in case of extreme emergency, are factors which may render this system undesirable in the smaller plants.

5. Combinations of 1 to 4 with economizers, air heaters or both. Among the large stations using the house turbine in connection with stage-bleeding and economizers may be mentioned the Weymouth Station of the Edison El. Ill. Co. of Boston, Northeast Station of the Kansas City Power Co., and the new units at Colfax.

In some of the latest installations, for example, the Edgar Station of

the Boston Edison Co., the auxiliaries are all motor-driven and the current is supplied by an auxiliary generator coupled directly to the main generator shaft. This makes each unit practically self-contained without taking auxiliary power from the main busbars. The auxiliary generator does not increase the electrical complications of the station, and the electric drives eliminate the auxiliary steam piping.

The various calculations in drawing up a heat balance differ in no way from those already shown in connection with condensers and feedwater heaters. The calculations in themselves are very simple but considerable judgment must be exercised in estimating the various quantities entering into the heat exchange. Thus, the value of the heat balance depends upon the accuracy attained in estimating (1) the water rates of the prime movers and auxiliaries at different loads, (2) coefficients of heat transfer in closed heaters and condensers of whatever type, (3) temperature depression between condensate and vapor entering the heater or condenser. (4) heat losses to the surroundings and (5) in drips. The usual procedure is to begin with the assumed conditions of the steam at the boiler nozzle and trace its path through the main unit, auxiliaries, heaters, and condensers, calculating the heat exchange as it passes through each piece of apparatus. At the same time the path of the condensate and makeup is followed from its source to the boiler. A complete mathematical treatise of a number of heat balances will be found in the following papers presented before the American Society of Mechanical Engineers.

Feed Heating for High Thermal Efficiency L. Holander, Trans. A.S.M.E., Vol. 44, 1922.

Economy Characteristics of Stage Feedwater Heating by Extraction: E. H. Brown and M. K. Drewry, Trans. A.S.M.E., Vol. 45, 1923.

Reheating in Central Stations: W. J. Wohlenberg, Trans A.S.M.E., Vol. 45, 1923.

High Pressure, Reheating, and Regenerating for Steam Power Plants: C. F. Hirshfeld and F. O. Ellenwood, Trans A.S.M.E., Vol. 45, 1923.

Power Station Heat Balance: Prime Movers Com. N.E.L.A., 1923, Part A, p. 51.

Heat Balance — Devon Station Power, Apr. 29, 1924, p. 671.

Relation of Auxiliary Drives to Heat Balance: Power, Dec. 6, 1921, p. 888.

Heat Balance at Colfax: C. W. Clarke, Trans. A.S.M.E., Vol. 43, p. 487.

Heat Balance at Delaware Station: E. L. Hoping, Trans A.S.M.E., Vol. 43, p. 481.

Heat Balance of Connors Creek Plant: C. H. Berry and F. E. Moreton, Trans. A.S.M.E., Vol. 43, p. 500.

PROBLEMS

1. Determine the amount of soda ash and lime necessary to soften 10,000 gallons of water as per analysis, Col. 2, Table 73.

2. In a certain plant it costs 30 cents per 1000 lb. to evaporate water from feed temperature of 60 deg. to steam at 115 lb abs. and 50 deg superheat; required the saving in per cent if the feedwater is heated by exhaust steam to 210 deg. fahr.

3. A 2000-kw. turbo-generator plant uses 18 lb. steam per kw-hr., initial pressure

140 lb. abs., back pressure 3 in. abs., superheat 100 deg. fahr., temperature of the condensate 100 deg. fahr.; auxiliaries develop 100 hp. and use 30 lb. steam per hp-hr. (non-condensing), initial pressure 115 lb. abs., steam dry at admission; required the temperature of the feedwater if the auxiliary exhaust is discharged into an open heater.

4. Required the tube surface necessary for a closed heater suitable for the conditions in Problem 3 Assume $U = 350$

5. If the tubes are $\frac{1}{2}$ in inside diameter, required the total length of water travel for the conditions in Problem 4, assuming a water velocity through the tubes of 120 ft. per min.

6. Raw water at a temperature of 200 deg. fahr. is pumped into the chamber of a flash-type evaporator in which a pressure of 7 in. abs. is maintained. What percentage of the raw water is flashed into vapor?

7. Determine the amount of raw water evaporated per lb. of steam supplied in a single-effect evaporator if the conditions are as follows: Pressure of steam supply, 17 lb. abs, vacuum, 21 in, temperature of raw water, 60 deg. fahr. Neglect all losses.

8. Same conditions as in Problem 7 except that a double-effect evaporator is to be used.

9. How many lb. of steam must be extracted from the 18 lb. abs stage of a 10,000-kw. steam turbine in order to heat the condensate to the maximum if the conditions are as follows. Initial pressure, 350 lb. gage, superheat, 250 deg. fahr., vacuum, 1 in. abs., water rate of turbine with full extraction at rated load 11 5 lb. per kw-hr., 10 per cent drop in pressure in steam to stage heater; temperature of condensate, 77 deg. fahr.

10. Given the following conditions for a single-stage extraction heater Initial pressure 265 lb. abs., superheat 150 deg. fahr., back pressure 1 in. abs., temperature of condensate 79 deg. fahr. Construct curves similar to those shown in Fig. 423 with a view of determining the temperature of the feedwater at which the efficiency of power generation is a maximum. Assume 100 per cent Rankine-cycle efficiency and neglect all heat losses

11. Calculate the final feedwater and flue-gas temperatures for a counterflow economizer installation operating under the following conditions. Boiler heating surface 10,000 sq. ft., economizer surface 6500 sq. ft., initial feedwater temperature 160 deg. fahr., initial flue-gas temperature 600 deg. fahr. when the boiler is operating at 150% above standard rating; coal used, Illinois washed nut, 13,500 B.t.u. per lb., $U = 4$.

CHAPTER XIV

PUMPS

266. Classification. — Pumps used in connection with steam power plants may be conveniently classified under five groups according to the principles of action.

1. **Piston pumps,** in which motion and pressure are imparted to the fluid by a reciprocating piston, plunger, or bucket. The action is positive and a certain definite amount of fluid is handled per stroke under predetermined conditions of pressure and velocity.

2. **Centrifugal pumps,** in which the fluid is given initial velocity and pressure by a rotating impeller. The action is not positive, as the amount of fluid discharged is not necessarily proportional to the impeller displacement.

3. **Positive-displacement rotary pumps,** in which motion and pressure are imparted to the fluid by a rotating impeller or screw. The volume discharged is practically equal to the impeller displacement regardless of pressure.

4. **Jet pumps,** in which velocity and pressure are imparted to the fluid by the momentum of a jet of similar or other fluid. The ordinary steam injector is the best known of this group.

5. **Direct-pressure pumps,** in which the pressure of one fluid acts directly on the surface of another fluid, thereby imparting all or part of its energy to the latter. The pulsometer is an example of this type.

These groups may be variously subdivided as follows:

Piston	Direct-acting	Simplex Duplex	Air. Vacuum. Forcing. Lifting.
	Flywheel Power-driven	Simplex Duplex Triplex	
Centrifugal	Volute Turbine.	Single-stage Multi-stage	Vacuum. Forcing. Lifting.
Rotary	Power-driven	Forcing Lifting	
Jet	Injector Ejector	Positive Automatic	
Direct Pressure	Pulsometer	Lifting	
	Air-lift	Lifting	

Piston, or plunger pumps, are adapted to widely diversified service. Boiler-feed, condensate, and vacuum pumps for small plants, city waterworks pumps, and high-pressure force pumps are ordinarily of this type. In the direct-acting type, Fig. 425, the water plunger and steam piston are secured to a single piston rod and the steam pressure is transmitted directly to the water. There is no flywheel, connecting rod, or crank. The volume of the delivery for constant initial steam pressure is inversely proportional to the resistance offered by the water; when the resistance equals the forward effort of the steam pressure, the pump stops. This

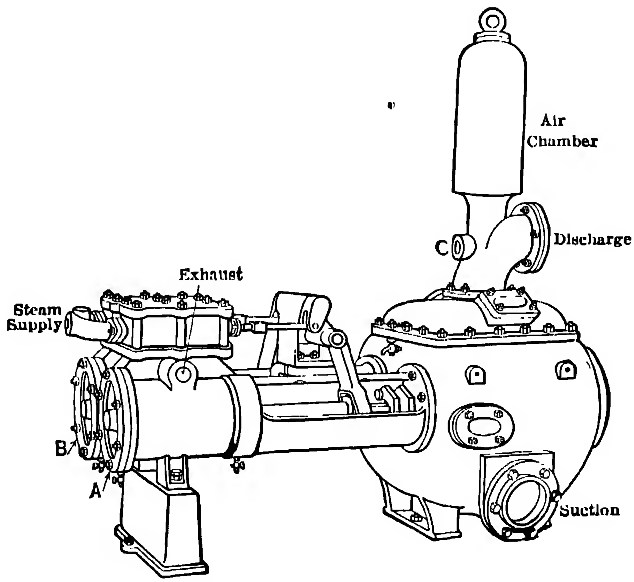


FIG. 424. Typical Duplex Pump

class of pump is well adapted for constant head and variable capacities, since it may be operated as slowly as suits the capacity requirements by simply throttling the discharge. The steam is not used expansively; therefore, the water rate is very large in proportion to the work performed.

Flywheel pumps, Fig. 443, are ordinarily classified as pumping engines. In this class steam may be used expansively, as sufficient energy is stored in a flywheel to permit the drop in steam pressure during expansion. These pumps find wide application in city waterworks, elevator plants, and the like, where high duty is required. They are little used as stationary boiler feeders, but are used to some extent in river-boat practice.

Piston pumps, Fig. 446, driven by gearing or belting, are ordinarily classified as power-driven pumps. The source of power may be a steam

engine, electric motor, or gas engine. The single-cylinder machine is often designated as a " simplex " power-driven pump, the two-cylinder as a " duplex," the three-cylinder as a " triplex," and so on.

Centrifugal pumps, Fig. 458, have largely supplanted the piston type for all but high-pressure low-capacity service because of their compactness, balanced rotary motion, absence of valves and pistons, uniform pressure and flow, freedom from shock, ability to handle dirty water, and high rotation speed permitting direct connection to electric motors and steam turbines. The mechanical efficiency of the centrifugal pumps is lower

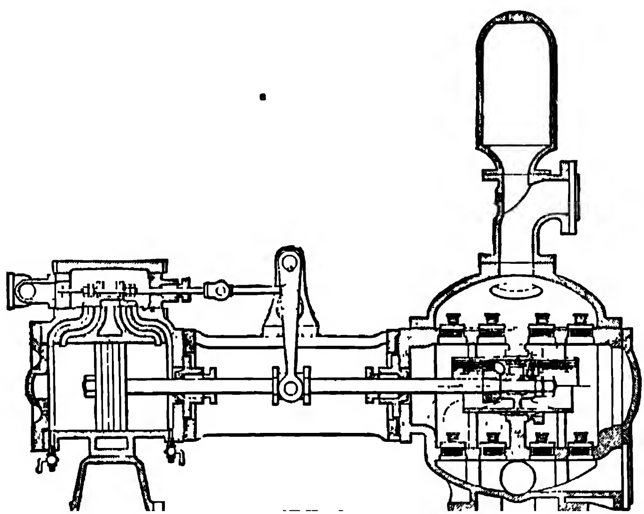


FIG. 425. Section Through a Typical Duplex Pump.

than that of the average piston pump, but this disadvantage is largely offset by low first cost and low maintenance costs.

Rotary pumps, Fig. 454, are employed to a limited extent in the same field as the centrifugal pump. Being positive in action, they permit of a much lower rotative speed for the same delivery pressure.

Jet pumps, Fig. 451, are seldom used as pumps in the ordinary sense of the word, on account of their extremely low efficiency, but are occasionally employed for discharging water from sumps. Their greatest field of application lies in boiler feeding, and in this connection their overall thermal efficiency is very high. Jet pumps of the steam-ejector type are much in evidence in the modern condensing plant for withdrawing or for assisting the main air pump in withdrawing the non-condensable gases and vapor entrainment from condensers. See paragraph 285.

Direct-pressure pumps operated by steam, such as the " pulsometer,"

Fig. 449, are used principally for pumping out sumps, surface drains, and the like, where the operation is intermittent. Direct-pressure pumps of the air-lift type, Fig. 485, are quite common and are used a great deal in situations where water is to be pumped from a number of scattered wells.

267. Direct-acting Steam Pumps. — Figure 424 shows a general assembly, and Fig. 425 a section, through one element of a typical direct-acting steam pump of the duplex type. This style of pump consists virtually of two single-cylinder pumps mounted side by side, the water ends and the steam ends working in parallel between inlet and exhaust pipe. The piston rod of one pump operates the steam valve of the other through the medium of bell cranks and rocker arms. The pistons move alternately, and one or the other is always in motion, the flow of water being practically continuous.

In general construction the steam pistons and valves are similar to those of steam engines. The valves in duplex pumps, however, have no lap. In order to reduce the valve travel to a minimum, and still have sufficient bearing surface between the steam ports and the main exhaust ports to prevent the leakage of steam from one to the other, separate exhaust ports are provided which enter the cylinder at nearly the same point as the steam ports. This arrangement offers a simple means of cushioning the piston by exhaust steam, thus preventing it from striking the cylinder heads at the ends of the stroke. The valves of the duplex pump, having no lap, would, if connected rigidly to the valve stem, open one port as soon as the other had been closed, at about mid-stroke of the piston, thus cutting down the stroke to about one-fourth the usual length. To obviate this difficulty the valves are given considerable lost motion by allowing sufficient clearance between the lock nuts on the valve stem; the latter, therefore, imparts no motion to the valve until the piston operating it has nearly completed the stroke. The lost motion between valves and lock nuts renders it impossible to stop the pump in any position from which it cannot be started by simply admitting steam, and therefore the pump has no dead centers. When one piston moves to the end of the stroke, it pulls or pushes the opposite valve to the end of its travel; then, when the piston starts back to the other end of its stroke, the valve remains stationary, owing to the lost motion, until the piston has completed about one-half the stroke. During this time the opposite piston has completed a full stroke and the valve operated by it will have opened the steam port wide, so that while one valve covers both steam ports the other is at the end of its travel. In some makes of pumps, the stem is rigidly attached to the valves, the lost motion being adjusted outside the steam chest, as shown in Figs. 426 and 427 which represent two common constructions of a duplex valve gear.

Figure 428 shows the valve and piston in the position occupied at the commencement of the stroke. At one end of the valve the steam port P is open wide, and at the opposite end the exhaust port E is open wide. When the piston nears the opposite end of the stroke and reaches the position shown in Fig. 429, the steam escape through the exhaust port E

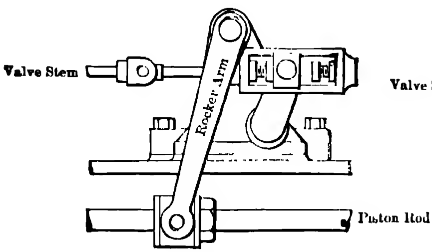


Fig. 426.

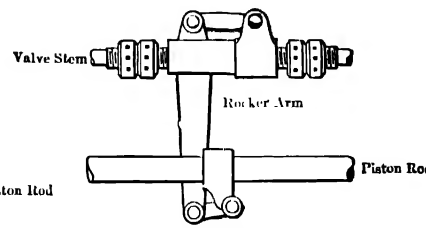


Fig. 427.

is cut off by the piston, and since the steam port is closed, the remaining steam is compressed between the piston and cylinder head, thus arresting the motion of the piston gradually without shock or jar.

The duplex direct-acting steam pump of the type just described is the most widely distributed device for general pumping service where delivering pressures do not exceed 200 lb. gage and capacities 1000 gal. per min.

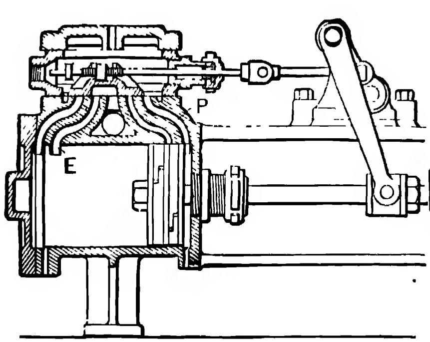


Fig. 428.

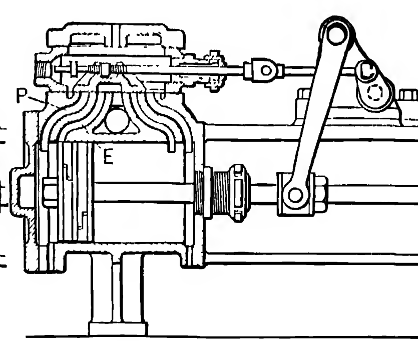


Fig. 429.

For tank supply and other similar light services, sizes as large as 4000 gal. per min. have been built, but the centrifugal pump is perhaps the better investment in this connection. The duplex steam pump owes its popularity to low first cost, low maintenance, simplicity of operation, and positive action, but the water rate is very high and "short-stroking" is common. Where it is desired to maintain the duplex principle and at the same time improve the water rate, the steam end is frequently compounded

as illustrated in Fig. 431. For pressures ranging from 200 to 1000 lb. gage or more, the water end is modified as shown in Fig. 433.

Single-cylinder or **simplex** direct-acting steam pumps are frequently preferred to the duplex type and are rapidly superseding the latter for general service. This style of pump differs from the duplex only in the construction of the steam end and in the employment of but one steam and one water cylinder. The older designs of simplex pumps were of the steam-actuated valve type in which the movements of the main steam valve, usually of the piston pattern, were controlled by a small pilot valve or valves. The steam supply to the pilot valve was controlled by the position of the main steam piston. This design is no longer in evidence except in the older plants, because the pilot valves are apt to stick and positive action of the pump cannot be depended upon. In the modern simplex designs the pilot valve is mechanically operated, and positive action is therefore insured. Among the many designs of simplex pumps may be mentioned the American-Marsh, Cameron, Knowles, Blake, Deane, Davidson, and Burnham. While these various designs differ widely in details of valve construction the basic principles are more or less alike. Figure 430 shows a sectional elevation through the steam end of

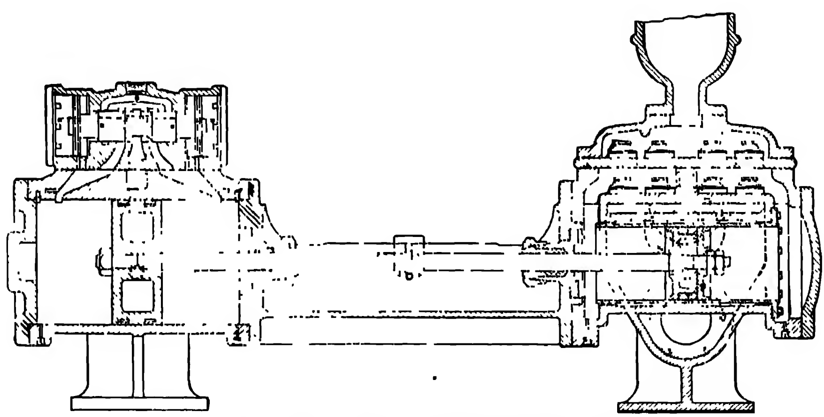


FIG. 430. American-Marsh Steam Pump.

the American-Marsh design and serves to illustrate the manner in which the reversal of the pump is effected at the end of the stroke. The pilot or auxiliary valve is of the semi-rotative disc type, actuated by a rocker-arm fastened to cross head on the main piston rod as indicated in the illustration. The movement of the auxiliary valve controls the steam supply to the main steam valve, which is of the balanced-piston type, and this in turn admits steam to and exhausts it from the steam cylinder. The length of the stroke is adjusted to suit varying conditions by means

of two regulating screws located at the side of the auxiliary valve stem lever. On pumps having a 10-in. stroke or longer, cushion valves are furnished at each end of the steam cylinder, which regulate the escape of exhaust steam and thereby control the cushioning effect. For a detailed description of the various types of Simplex pumps, consult "Pumping Machinery," by Arthur M. Greene, John Wiley & Sons, Inc., Pub. The sim-

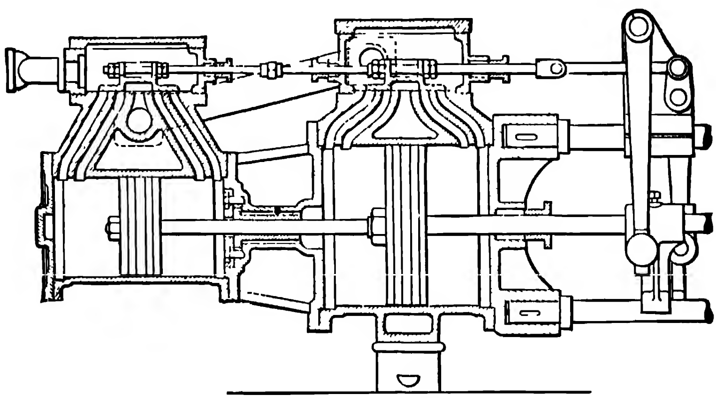


FIG. 431. Typical Compound Duplex Pump.

plex pump valve-control mechanism is a little more complex than the duplex, but with it there is little danger of short-stroking. Water rates are lower than with the duplex type because of the reduced clearance volume.

268. Pump Valves. — In the large majority of pumps for pressures under 200 lb. gage, it is general practice to use a number of small valves of the spring-loaded flat-disc type. The disc packing is renewable and is composed of various materials, depending upon the temperature and character of the fluid to be pumped. This packing is usually composed of soft-rubber compounds for cold water and of hard rubber, compressed fiber, or special metallic alloy for hot water. The discs are held firmly to the seat by conical or spiral springs and guided by a bolt through the center, as illustrated in Fig. 432. For pressures over 200 lb. gage, it is customary to use a comparatively small number of spring-loaded metallic valves, wing-guided from below and working in bronze seats, as shown in Fig. 433. The valve chambers are comparatively small castings, separate from the cylinder and connected together by branch manifolds for the suction and discharge pipes. The valve-pot covers form the guides for

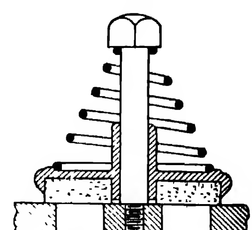


FIG. 432. A Typical Pump Disk-Valve.

the valve springs. Each valve chamber may be opened up for inspection or repair without disturbing any other part of the water end. In some designs of pumping engines, large mechanically operated valves are used on the water end, a single suction and a single delivery valve for each

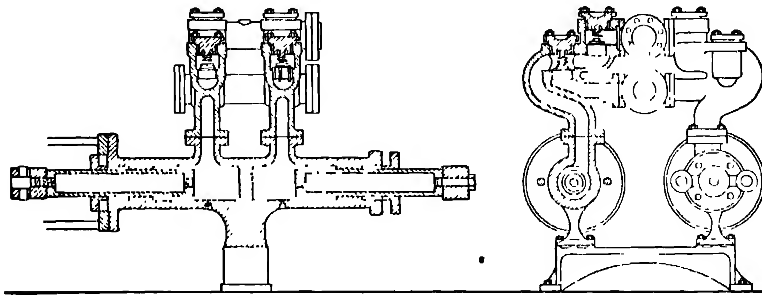


FIG. 433. Worthington Pot-valve Pressure Pump.

end of the cylinder, in place of a number of spring-loaded disc valves. The Riedler pump, Fig. 434, is of this design.

269. Air and Vacuum Chambers. — Air chambers in piston pumps are for the purpose of causing a steady discharge of water and of reducing excessive pounding at high speeds by providing a cushion for the water. The water discharged under pressure compresses the air in the air chamber somewhat above the normal pressure of discharge during each stroke of the water piston, and when the piston stops momentarily at the end of the stroke the air expands to a certain extent and tends to produce a uniform rate of flow.

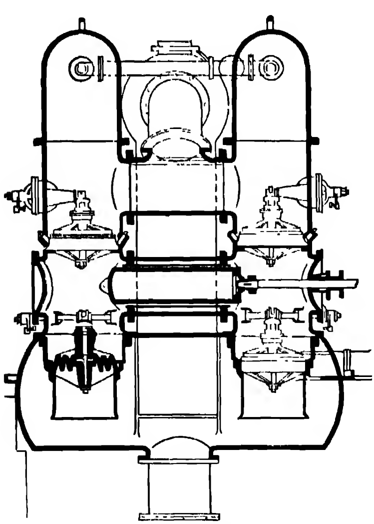


FIG. 434. Riedler Mechanically Operated Pump Valves.

The volume of the air chamber varies from 2 to 3 1/2 times the volume of the water piston displacement in single-cylinder pumps, and from 1 to 2 1/2 times in the duplex type. High-speed pumps are provided with air chambers of from 5 to 6 times the piston displacement. The water level in the air chamber should be kept down to one-fourth the height of the chamber. In slow-running pumps, sufficient air may be carried into the pump chamber along with the water, but, with high speeds, a large part of the

air will be discharged, and air must be forced into the chamber by mechanical means. The larger the chamber, the more uniform will be the discharge pressure.

Vacuum chambers are frequently provided for the purpose of maintaining a uniform flow of water in the suction pipe and assisting in the reduction of slip. Such chambers should be of slightly greater volume than the suction pipe and of considerable length rather than diameter. Figure 435 illustrates two designs commonly used. The one in Fig. 435 (*B*) should be placed in such a position as to receive the impact of the column of water in the suction pipe as illustrated in Fig. 436 (*A*), (*B*), and (*C*). The chamber illustrated in Fig. 435 (*A*) should be placed in the suction pipe below but close to the pump.

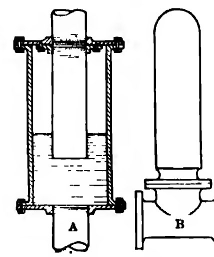


Fig. 435. Forms of Vacuum Chambers.

270. Water Pistons and Plungers. — In cold-water pumps the water pistons are usually packed with some kind of soft packing. Figure 437 (*A*) shows the details of a piston with square **hydraulic packing.** The body *E* is fastened to the piston rod by nut *C;* packing is placed at *D,* and follower *F* is forced up by the nut *B* and locked

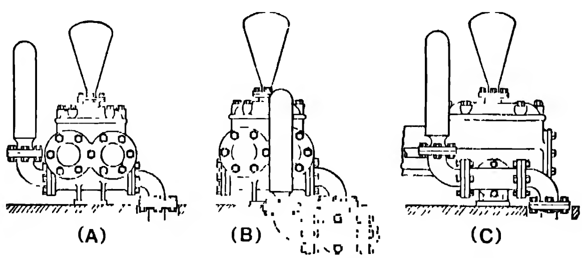


Fig. 436. Different Arrangements of Vacuum Chambers.

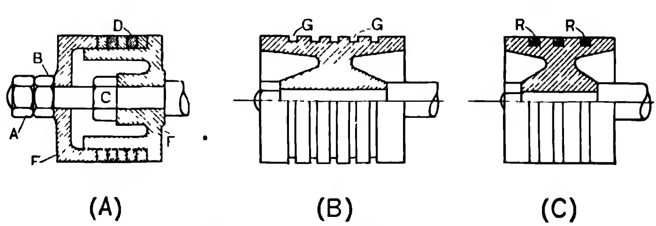


Fig. 437. Types of Water Pistons.

by nut *A*. For large sizes the design is the same except that the follower is set up by a number of nuts near the edge. In hot-water pumps the pistons are often packed by means of **metallic piston rings,** *R, R,* **Fig.**

437 (*C*), similar to those in steam pistons, or merely by **water grooves,** ***G, G,*** Fig. 437 (*B*).

The **cup-leather,** Fig. 438, is a good packing for single-acting deep-well and high-pressure cold-water pumps, since the pressure exerted by the leather varies with the water pressure. A cup-leather packing for a double-acting pump is shown in Fig. 439. For very heavy pressures, the **U-leather** packing, Fig. 440, has given the best satisfaction.

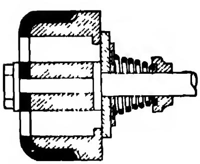


FIG. 438. Cup-leather Packing.

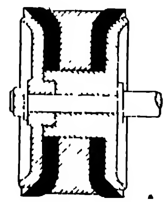


FIG. 439. Double-leather Packing.

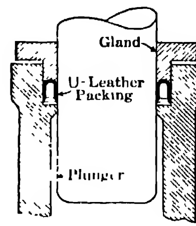


FIG. 440. U-leather Packing.

The water end is often fitted with a **plunger** instead of a piston, as in Figs. 441 and 442. The piston is more compact, but the plungers do not require a bored cylinder, so that the first cost is not materially different.

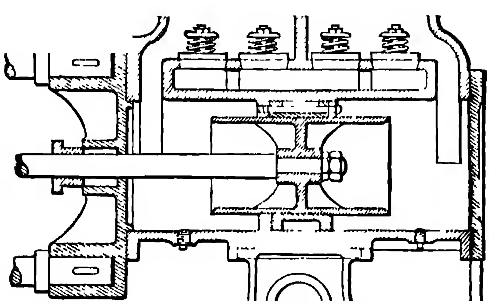


FIG. 441. Plunger with Metal Packing Ring.

Figure 441 shows a plunger with metal packing ring. When leakage becomes excessive it is necessary to renew the ring, which is readily removed.

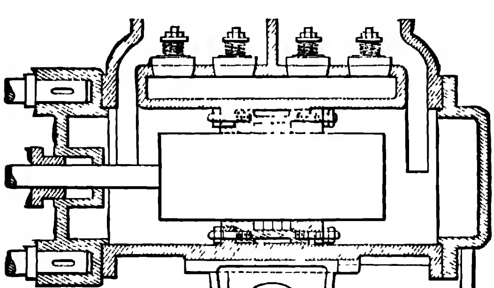


FIG. 442. Plunger with Hydraulic Packing Ring.

In Fig. 442 the plunger is packed with the hydraulic packing, as in the follower type of pump piston. The great difficulty with the above types of piston and plunger is in keeping the packing tight or in knowing when it is leaking, and the trouble necessary to replace the packing. The **outside-packed plunger,** Fig. 433, obviates these disadvantages to a **great**

extent, since leakage is readily detected and repacking is performed without removing the cylinder heads. In dirty or dusty locations, however, the piston pump or inside-packed plunger is to be preferred, since the abrasive action of the dust renders outside packing difficult. Figure 443 illustrates a high-duty elevator pump with outside-packed plunger.

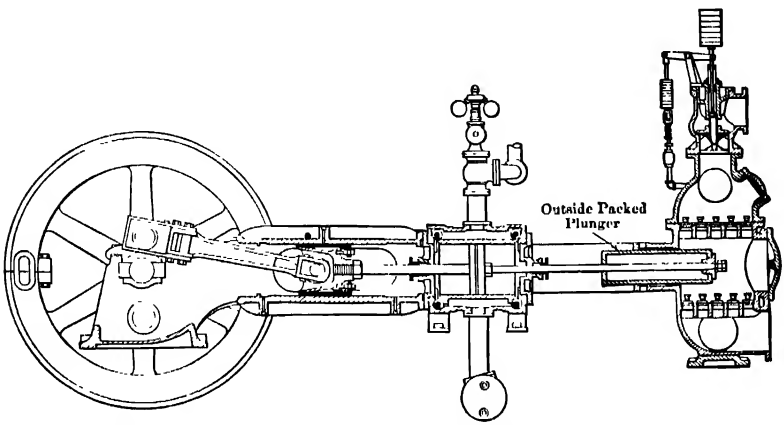


FIG. 443. Horizontal Flywheel Pump with Outside-packed Plunger.

271. Performance of Piston Pumps. — The mechanical efficiency of the water end of a piston pump is very high, since there is very little friction to overcome. Tests conducted on direct-acting steam pumps show mechanical efficiencies (water-end i.hp. divided by steam-end i.hp.) ranging from 90 to 99 per cent. With very small pumps operating against low heads, the efficiency may be as low as 50 per cent because the ratio of the friction of the packing and stuffing box to total pressure acting on the piston increases with the decrease in head. Overall mechanical efficiencies (actual water hp. delivered divided by steam-end i.hp.) are considerably less and vary within wide limits because of short-stroking and slip past piston and valves. Pump slip varies from 2 to 40 per cent or more, depending upon the design and condition of the pump and valves and the number of strokes; an average value for piston and plunger pumps in first-class condition is 8 per cent when operating at rated capacity. However, to allow for possible short-stroking and leakage caused by wear, it is customary to make a liberal allowance and the actual delivery is usually assumed to be about 85 per cent of the piston displacement. Direct-acting pumps as a class are very wasteful of fuel and low in thermal efficiency, largely because of the non-expansive use of steam and slow-speed operation. Rankine-cycle ratios at rated capacity vary from 15 per cent in the smallest sizes to about 50 per cent in the larger units, corresponding

to water rates of approximately 250 to 72 lb. per i.hp-hr. for the usual saturated steam conditions. The simplex design, because of the lower clearance volumes and better steam distribution, is from 10 to 20 per cent more economical in the use of steam than the duplex of equal capacity. The average single-expansion direct-acting boiler-feed pump, when operating under service conditions, uses approximately 5 per cent of the boiler steam; but if the pump exhaust is utilized in heating the feedwater, the net heat consumption is somewhat less than 1 per cent. Compound direct-acting pumps running non-condensing use from 40 to 100 lb. of steam per i.hp-hr., depending upon the steam conditions and the ratios of the steam-cylinder diameters. Single-cylinder flywheel pumps of the slow-speed type, running non-condensing, use about 50 lb. of steam per i.hp-hr. Multi-cylinder flywheel pumps of the high-duty type use about 25 lb. per i.hp-hr. when running non-condensing, and as low as 10 lb. when operating condensing. High-grade direct-connected motor-driven

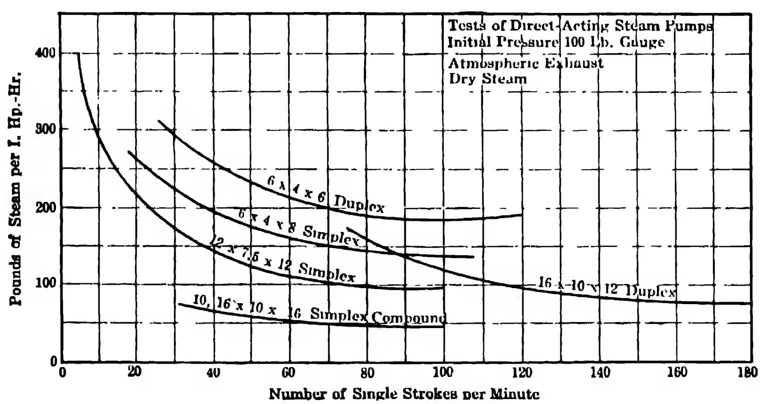


Fig. 444.

power pumps have a mechanical efficiency from line to water load, at normal rating, of about 80 per cent. The efficiency of *geared* pumps at normal rating varies with the character of the gearing and the degree of speed reduction, and may range anywhere from 25 to 75 per cent. The steam consumption of all direct-acting steam pumps appears to decrease with the increase in speed, as shown in Fig. 444, the curves of which were plotted from a number of scattering tests.

Nearly all types of direct-acting piston pumps can be operated with a moderate amount of superheat before serious operating difficulties arise, the exact amount, of course, depending upon the design and construction. With 25 deg. fahr. of superheat the water rate of the single-expansion

simplex or duplex pump may be reduced from 10 to 15 per cent, and with 50 deg. superheat from 15 to 20 per cent.

Example 69. — A small direct-acting duplex pump uses 150 lb. of steam per i.hp-hr. Gage pressure 150 lb. per sq. in., feedwater temperature 64 deg. fahr. Required the per cent of rated boiler capacity necessary to operate the pump.

Solution. — The steam pressure pumped against, 150 lb. per sq. in., is equivalent to $150 \times 2.3 = 345$ ft. of water. The friction through the valves, fittings, and pipe, and the vertical distance between suction and feedwater inlet, are assumed to be equivalent to 20 per cent of the boiler pressure, giving a total head of $150 + 30 = 180$ lb. per sq. in. or 414 ft. of water.

A boiler horsepower, taking into consideration leakage losses and the steam used by the feed pump, will be equivalent to the evaporation of approximately 32 lb. of water per hr. from a feed temperature of 64 deg. fahr. to steam at 150 lb. gage.

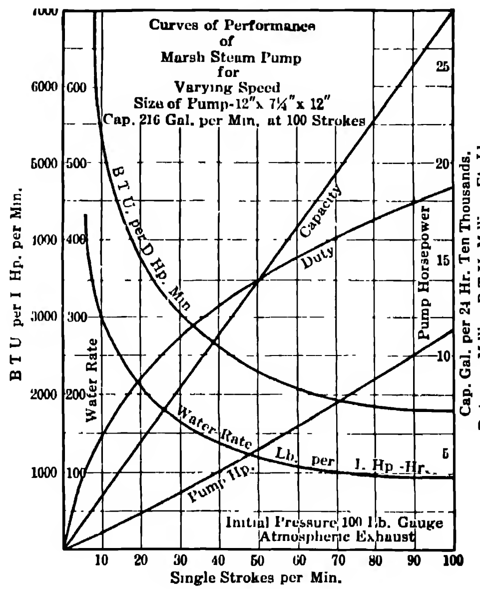


FIG. 445.

The actual work done in pumping 32 lb. of water against a head of 414 ft. is

$$414 \times 32 = 13{,}248 \text{ ft-lb.}$$

This corresponds to

$$13{,}248 \div 60 \times 33{,}000 = 0.0067 \text{ hp.}$$

The total heat of 1 lb. of steam above 64 deg. fahr. is 1163 B.t.u. The heat delivered to the pump per i.hp-hr. is

$$1163 \times 150 = 174{,}450 \text{ B.t.u.}$$

The amount used by the pump for each boiler horsepower, disregarding efficiency, is

$$174{,}450 \times 0.0067 = 1168 \text{ B.t.u. per hr.}$$

The overall mechanical efficiency of the average feed pump ranges from 50 to 85 per cent, depending upon its condition and the number of strokes per min. Assuming it to be 65 per cent, the heat used by the pump per hr. to deliver 32 lb. of water into the boiler is

$$1168 \div 0.65 = 1796 \text{ B.t.u.}$$

A boiler horsepower is equivalent to 33,479 B.t.u. per hr. Therefore the per cent of boiler output necessary to operate the pump is

$$100 \times 1796 \div 33{,}479 = 5.36 \text{ per cent.}$$

If the exhaust steam is used for heating the feedwater, the steam consumption will be 0.73 per cent of the boiler capacity. Thus, the weight of steam consumed per boiler hp-hr.

$$1796/1163 = 1.54 \text{ lb.}$$

Allowing a 10 per cent loss, the heat in the exhaust available for heating the feedwater is

$$[1150 - (64 - 32)]\, 0.9 \times 1.54 = 1550 \text{ B.t.u.}$$

1796 − 1550 = 246 B.t.u., or the net heat required by the pump per hr. to deliver 32 lb. of water to the boiler.

The per cent of boiler output necessary to operate the pump is

$$100 \times 246/33{,}479 = 0.73.$$

Pump performances are generally given in terms of the foot-pounds of work done by the water piston per thousand lb. of dry steam or per million B.t.u. consumed by the engine, thus:

$$1.\ \text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Weight of dry steam used}} \times 1000 \qquad (251)$$

$$2.\ \text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Total number of heat units consumed}} \times 1{,}000{,}000 \qquad (252)$$

(See A.S.M.E. Code for conducting duty trials of pumping engines, *Trans. A.S.M.E.*, Vol. 37, 1915.)

Example 70. — A compound feed pump used 100 lb. of steam per i.hp-hr.; i.hp., 48; capacity, 400 gal. per min.; temperature of water, 200 deg. fahr.; total head pumped against, 175 lb. per sq. in.; steam pressure, 100 lb. gage; moisture in the steam, 3 per cent. Required the duty on the dry steam and on the heat-unit basis.

Solution. — 175 lb. per sq. in. is equivalent to 175 × 2.4 = 420 ft. of water at 200 deg. fahr.

Weight of 400 gal. of water at 200 deg. fahr. = 400 × 8.03 = 3212 lb.

Work done per min. = 3212 × 420 = 1,349,040 ft-lb.

Weight of dry steam supplied per min.

$$= \frac{100 \times 48}{60} \times 0.97 = 77.6 \text{ lb.}$$

B.t.u. supplied per min.

$$= \frac{100 \times 48}{60} (0.97 \times 879.8 + 309 - 200 + 32) = 79{,}552.$$

Duty per thousand lb. of dry steam

$$= \frac{1,349,040}{77.6} \times 1000 = 17,384,150 \text{ ft-lb.}$$

Duty per million B.t.u.

$$= \frac{1,349,040}{79,552} \times 1,000,000 = 16,958,000 \text{ ft-lb.}$$

Table 85 may be used in approximating the duty, thus:

The mechanical efficiency of the pump in the preceding problem is

$$\text{Efficiency} = \frac{\text{water hp.}}{\text{i.hp.}} = \frac{1,349,040}{33,000 \times 48} = 85 \text{ per cent.}$$

At the intersection of vertical column "85" and horizontal column "100" of Table 82, we find 16.82 millions. See, also, Table 63.

272. Size of Direct-acting Steam Pumps. — Let

D = diameter of water cylinder, in.
d = diameter of the steam cylinder, in.
L = length of stroke, in.
N = number of working strokes per min.
H = head in feet between suction and boiler water level.
R = resistance in lb. per sq. in. between suction level and boiler water level due to valves, pipes, and fittings.
p = boiler pressure, lb. per sq. in.
p' = steam pressure on the piston, lb. per sq. in.
C = ft. of water equivalent to one lb. per sq. in. pressure.
S = ratio of the water actually delivered to the piston displacement.
W = weight of water delivered, lb. per hr.
I = indicated horsepower of the pump at maximum capacity.
E = mechanical efficiency of the pump, taken as the ratio of the br.hp. at the discharge opening to the i.hp. of the pump, steam end.

Then

$$W = \frac{\pi}{4} \cdot \frac{D^2}{144} \cdot \frac{LN}{12} \times 60 \times 62.5 \times S = 1.7\, D^2LNS \tag{253}$$

$$D = 0.77 \sqrt{W/LNS}. \tag{254}$$

$$d = D \sqrt{(p + R + H \div C)/p'} \tag{255}$$

$$I = W\,[(p + R)\,C + H)] \div (33,000 \times 60 \times E) \tag{256}$$

In average practice the piston or plunger displacement is made about twice the capacity found by calculation from the maximum amount of water required for the engine, to allow for leakage, steam consumption of the auxiliaries, and blowing off.

TABLE 85

PERFORMANCE OF STEAM PUMPS. DUTY IN MILLIONS OF FT-LBS. PER MILLION B.T.U.

Steam Consumption, Lb. per I.hp-hr.		Mechanical efficiency = $\frac{\text{Pounds discharged per min.} \times \text{head in feet}}{\text{I.hp of steam cylinder} \times 33{,}000}$									
		0.95	0.90	0.85	0.80	0.75	0.70	0.65	0.60	0.55	0.50
Initial Press. 100 Lb. Gage, Non-Cond.	200	9.40	8.91	8.42	7.92	7.42	6.93	6.44	5.94	5.45	4.95
	190	9.90	9.39	8.86	8.36	7.83	7.31	6.79	6.27	5.74	5.22
	180	10.45	9.90	9.38	8.80	8.25	7.70	7.15	6.60	6.05	5.50
	170	10.90	10.50	9.90	9.32	8.74	8.15	7.57	6.99	6.41	5.83
	160	11.75	11.13	10.51	9.90	9.28	8.66	8.04	7.42	6.81	6.19
	150	12.55	11.90	11.22	10.59	9.90	9.29	8.49	7.92	7.26	6.60
	140	13.45	12.75	12.02	11.32	10.61	9.90	9.20	8.49	7.76	7.07
	130	14.49	13.71	12.96	12.20	11.42	10.69	9.90	9.15	8.38	7.62
	120	15.67	14.85	14.03	13.20	12.38	11.55	10.71	9.90	9.08	8.25
	110	17.10	16.21	15.31	14.40	13.50	12.60	11.70	10.80	9.90	9.00
	100	18.81	17.82	16.82	15.82	14.84	13.86	12.88	11.88	10.89	9.90
	90	20.90	19.80	17.76	16.72	15.67	14.63	13.58	12.54	11.49	10.45
	80	23.51	22.27	21.03	19.80	18.56	17.32	16.08	14.85	13.61	12.37
	70	26.90	24.50	24.04	22.64	21.22	19.80	18.40	16.98	15.58	14.14
	60	31.35	29.70	28.05	26.40	24.75	23.10	21.43	19.80	18.15	16.50
	50	37.62	35.64	33.64	31.68	29.68	27.72	25.76	23.76	21.98	19.80
	40	47.02	44.55	42.07	39.60	37.12	34.65	32.17	29.70	27.22	24.75
	30	62.70	59.40	56.10	52.80	49.50	46.20	42.86	39.60	36.30	33.00
Initial Press. 150 Lb. G. Vacuum 2 Lb. Abs	25	68.40	64.80	61.20	57.60	54.00	50.04	46.80	43.20	39.60	36.00
	20	85.50	81.00	76.50	72.00	67.50	63.00	58.50	54.00	49.50	45.00
	18	95.00	90.00	85.00	80.00	75.00	70.00	65.00	60.00	55.00	50.00
	16	106.87	101.25	95.62	90.00	84.37	78.75	73.12	67.50	61.87	56.25
	15	114.00	108.00	102.00	96.00	90.00	84.00	78.00	72.00	66.00	60.00
	14	122.14	115.71	109.29	102.86	96.43	90.00	83.57	77.14	70.71	64.28
	13	131.53	124.61	117.61	110.77	103.84	96.92	90.00	83.07	76.15	69.22
	12	142.50	135.00	127.50	120.00	112.50	105.00	97.50	90.00	82.50	75.00
	11	155.45	147.30	139.00	130.90	122.70	114.50	106.30	98.16	90.00	81.80
	10	171.00	162.00	153.00	144.00	135.00	126.00	117.00	108.00	99.00	90.00

For pumps with strokes of 12 in. or over, the speed of the plunger or piston is usually limited to 100 ft. per min. as a maximum, to insure smooth running. For shorter strokes a lower limit should be used. The maximum number of strokes ranges from 100 for strokes over 12 in. in length to 200 for strokes under 5 in. Boiler-feed pumps should be designed to give the desired capacity at about one-half the maximum number of strokes or less.

TABLE 86

MAXIMUM HEIGHT TO WHICH A PUMP CAN LIFT WATER BY SUCTION AT DIFFERENT TEMPERATURES

(Barometer 29.92)

Temperature, Deg. Fahr.	Maximum Lift, Ft		
	Theoretical	Favorable Conditions	Average Conditions
40	33.6	28	25
50	33.5	27	24
60	33.4	26	23
70	33.1	25	20
80	32.8	23	18
90	32.4	21	16
100	31.9	19	13
110	31.2	17	11
120	30.3	14	9
130	29.2	12	6
140	27.8	10	4
150	25.4	7	2
160	23.5	5	0
170	20.3	2	*2
180	16.7	*1	*5
190	12.8	*3	*7
200	7.6	*5	*9
210	1.3	*8	*11

* Pressure head.

Pump slip varies from 2 to 40 per cent, depending upon the condition of the piston and valves and the number of strokes. An average value for piston and plunger pumps in first-class condition is 8 per cent when operating at rated capacity, but it is wise to allow a much larger figure, say 20 per cent, for leakage caused by wear.

The area of the steam cylinder of a boiler-feed pump ranges from two to three times that of the water end, to allow for the various friction losses and to permit the pump to operate at reduced steam pressures. The total head pumped against includes the suction lift, friction of valves and fittings, the distance between the suction inlet and the boiler level, and the boiler pressure. The total head ranges in practice from 1.1 to

1.5 times the pressure in the boiler. When specific data are not available, the factor is ordinarily taken as 1.25. The application of equations (253) to (256), including the practical considerations stated above, is best illustrated by a specific example.

Example 71. — Calculate the size of direct-acting single-cylinder boiler-feed pump necessary to supply water to 1000 hp. of boilers operating at rated capacity. Gage pressure 100 lb. per sq. in., feedwater temperature 150 deg. fahr.

Solution. — One boiler hp-hr. is equivalent to 34.5 lb. water from and at 212 deg. fahr. or 31.2 lb. from a feedwater temperature of 150 deg. fahr., to dry steam at 100 lb. gage, therefore

$$W = 31.2 \times 1000 = 31{,}200 \text{ lb. per hr.}$$

To allow for wear and leakage, assume $S \fallingdotseq 0.80$.

Taking the maximum piston speed as 100 ft. per min. and assuming that the rated capacity is to be furnished with the pump operating at half the maximum speed, we have

$$LN = 100 \times 12 \div 2 = 600 \text{ in. per min.}$$

Substituting these values in equation (254) and reducing

$$D = 0.77 \sqrt{31{,}200/(600 \times 0.8)} = 6.2 \text{ in.}$$ — call it 6 in. since the assumptions have been liberal.

Assume the total head to be 1.25 p, i.e. $p + R + H \div C = 1.25\ p$, $p' = 0.50\ p$ and $E = 0.65$.

Substituting these values in equation (255) and reducing

$$d = 6 \sqrt{1.25 \times 100/(0.50 \times 100)} = 9.5 \text{ in., or, say, 10 in.}$$

On a basis of 100 strokes per min. as the maximum speed,

$$L = LN \div 100 = 1200 \div 100 = 12 \text{ in.}$$

The dimensions of the pump, therefore, will be 10 × 6 × 12.

The i.hp. at rated load may be obtained by substituting the proper values in equation (256), thus:

$$I = \frac{31{,}200 \times 1.25\ (100) \times 2.35}{33{,}000 \times 60 \times 0.65} = 7.1 \text{ i.hp.}$$

273. Power Pumps — Piston Type. — Piston pumps, geared, belted, or direct-connected to electric motors, gas engines, and water motors, are used chiefly in industrial plants. Their general utility is evidenced by the rapidly increasing number installed in situations formerly occupied by the direct-acting steam pump. The efficiency of this type of pump depends in a large measure upon the character of the driving motor and the efficiency of the transmitting mechanism. High-speed power pumps direct-connected to electric motors give efficiencies from line to water

horsepower as high as 83 per cent, while the low-speed geared type seldom exceeds 70 per cent. The curves in Fig. 447 give the performance of a direct-connected triplex pump, and those in Fig. 448 the performance of a triplex pump geared to an electric motor. Both of these performances are exceptionally good and are considerably above the average.

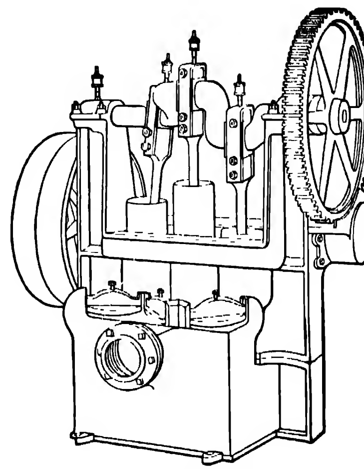


Fig. 446. A Typical Geared Triplex Pump.

The torque required to start plunger pumps may range from 125 to 250 per cent of normal full torque, depending primarily upon the tightness of the stuffing boxes and fit of pistons or plungers. Large pumps are frequently equipped with a by-pass for relieving the pressure during the starting period, thereby decreasing the initial torque required from the driving motor. With direct-current supply, compound motors having about 20 per cent series winding are recommended for this class of pumps, for both constant and variable speed operation. A compound motor of this design will provide the necessary starting torque without excess inrush of current. With alternating-current supply, squirrel-cage motors are permissible up to and including 5 hp. provided the power company will permit motors of this capacity to be thrown directly on the line. For larger capacities, wound-rotor motors of either the slip-ring or automatic self-starting type should be used since they develop the required initial torque without excessive line current. The squirrel-cage type of motor is inherently a constant-

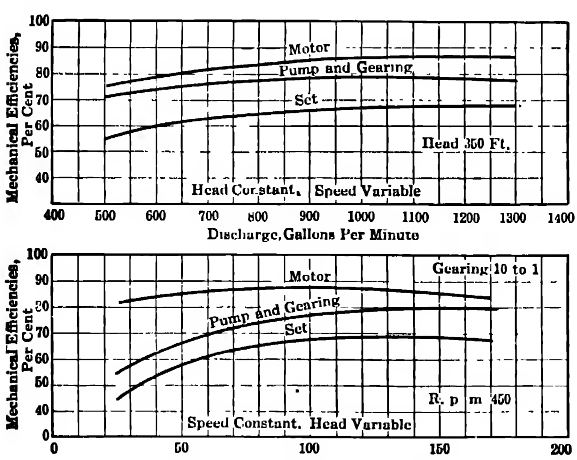


Fig. 447. Performance of a 65-hp. Motor-driven Triplex Pump. Geared Type.

speed machine and there is no satisfactory way of adjusting its speed. While speed adjustments are obtainable over a considerable range with a wound-rotor motor having resistance connected in series with the rotor, the number of operating speeds is limited to the number of points in the controller. For variable-speed service, the brush-shifting type of a.c. motor is finding favor with many engineers because it has starting characteristics similar to those of the slip-ring type and at the same time the efficiency is better at reduced speeds.

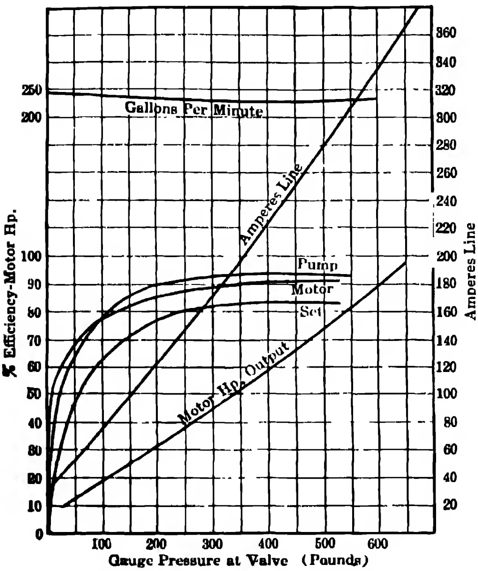


FIG. 448. Performance of a Motor-driven Triplex Pump. Direct Connected.

274. The Pulsometer. — This pump, representative of the direct steam-pressure type, is frequently used where comparatively small quantities of water (2000 gal. per min. and under) are to be lifted to moderate heads (75 ft. and under) and where low first cost, portability and simplicity of installation are essential factors. Pulsometers are adapted to the pumping out of sumps where the water is dirty and gritty, and to the handling of paper-mill pulp and the like. The steam consumption is approximately that of a duplex steam pump of equal capacity. Referring to Fig. 449, the operation is as follows: The pump is first primed by pouring water into the vessel through a plugged opening. Steam is then admitted into the right-hand chamber, which forces the water through the discharge valve by direct pressure. The moment the water falls to the level of the opening leading to the discharge chamber, the even surface of the water is broken up, and, owing to the peculiar form of the pump chambers, the water and steam are thoroughly churned up and brought into intimate contact, causing instant condensation of the steam. This creates a partial vacuum in the chamber, pulls the ball valve over the inlet opening, and shuts off the steam. Water then flows through the suction

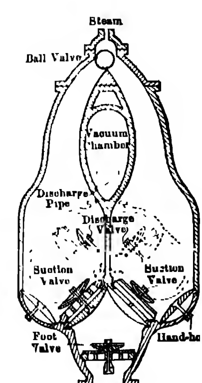


FIG. 449. The Pulsometer.

opening into the vacuum chamber. While the right-hand chamber is filling up, the left-hand chamber is emptying. The cycle continues in this manner as long as the pump is supplied with steam and water.

275. Injectors. — As a boiler feeder, the injector is an efficient and convenient device, cheap and compact, with no moving parts; it delivers hot water to the boiler without preheating, and has no exhaust steam to be disposed of. Its adoption in locomotives is practically universal, but in stationary practice it is limited to small boilers or single boilers or as a reserve feeder in connection with pumps. The objections to an injector are its inability to handle hot water, the difficulty of maintaining a continuous flow under extreme variation of load, and the uncertainty of operation under certain conditions. Figure 450 illustrates the simplest

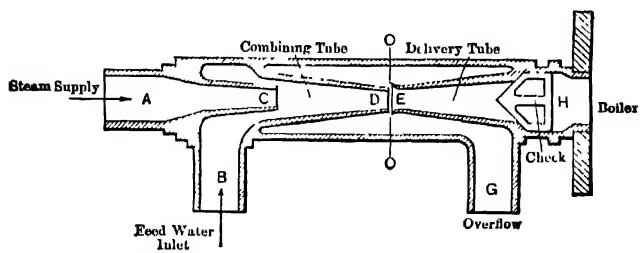


FIG. 450 Elementary Steam Injector.

form of single-tube injector. Boiler steam is admitted at A and, flowing through nozzle and combining tube to the atmosphere through G, partially exhausts the air from pipe B, thereby causing the water to rise until it comes in contact with the steam. The steam emerging from nozzle C at high velocity condenses on meeting the water and imparts considerable momentum to it. The energy in the rapidly moving mass is sufficient to carry it across opening O, lift check H from its seat, and force it into the boiler. The steam then ceases to escape at G.

Positive Injectors. — Figure 451 shows a section through a Hancock injector, illustrating the principles of the double-tube positive type. Its operation is as follows: Overflow valves D and F are opened and steam is admitted, which at first passes freely through the overflow to the atmosphere and in so doing exhausts the air from the suction pipe. This causes the feedwater to rise until it meets the jet of steam, and the two are forced through the overflow. As soon as water appears at the overflow, valve D is closed, valve C partially opened, and valve F closed. This admits steam through the forcing jet W and, the overflow valves being closed, the water is fed into the boiler. In case the action is interrupted for any reason, it is necessary to restart it by hand.

The chief advantage of the double-tube positive type lies in its ability to lift water to a greater height and to handle hotter water than the single-tube. Its range in pressure is also greater, that is, it will start with a lower steam pressure and discharge against a higher back pressure. Double-tube injectors are used almost exclusively in locomotive work.

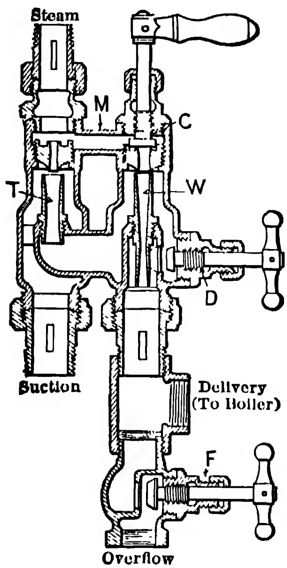


FIG. 451. Hancock Double-tube Injector.

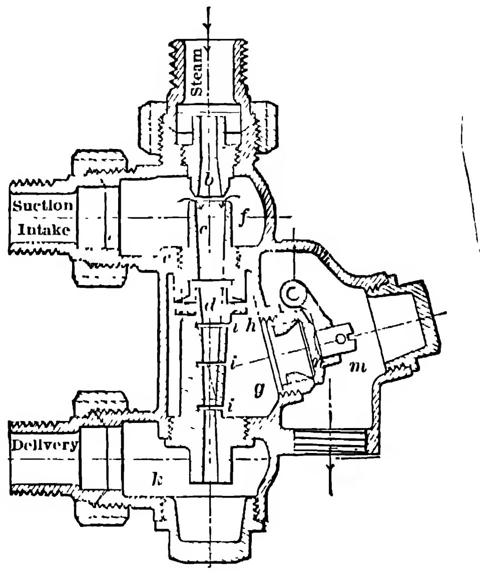


FIG. 452. Penberthy Automatic Injector.

Automatic Injectors. — Figure 452 shows a section through the Penberthy injector. Its operation is as follows: Steam enters at the top connection and flows through suction tube *c* into the combining tube *d* and into chamber *g*, from which it passes through overflow valve *n* to the overflow *m*. When water is drawn in from the suction intake and begins to discharge at the overflow, the resulting condensation of the steam creates a partial vacuum above the movable ring *h*, and the latter is forced against the end of tube *c*, cutting off the direct flow of water to the overflow. The water then passes into the boiler. Spill holes *i*, *i*, *i* are for the purpose of relieving the excess of water until communication with the boiler has been established. The action of opening and closing the overflow is entirely automatic. Where the conditions are not too extreme, the automatic injector is to be preferred for stationary work because of its restarting features. It is also used on traction, logging, and road engines, where its certainty of action and special adaptability render it invaluable for the rough work to which such machines are subjected.

Injectors, Theory of: Trans. A.S.M.E., 10–339; Sibley Jour., Dec., 1897, p. 101; Power, May, 1901, p. 23; Theory of the Steam Injector, Kneass.

Injectors, General Description: Power, Mar. 21, 1922, p. 160.

276. Performance of Injectors.—Since the heat given up by the steam must be equal to that absorbed by the feedwater, plus the heat equivalent of the work done in forcing the water into the boiler, plus any heat loss to the surroundings, the performance of an injector may be calculated from the relationship

$$H - q_2 = w\,(q_2 - q_o) + [(w + 1)h_1 + wh_o + F)]/778 \qquad (257)$$

in which

H = heat content of the steam supplied to injector, B.t.u. per lb. above 32 deg. fahr.

q_2 = heat content of the water discharged from injector, B.t.u. per lb.

q_o = initial heat content feedwater temperature, B.t.u. per lb.

w = lb. of water delivered per lb. of steam supplied.

h_1 = boiler pressure expressed in ft. of water.

h_o = suction lift, ft.

F = friction and loss to surroundings, ft-lb.

For all power plant purposes, it is sufficiently accurate to neglect the quantity in brackets and to assume that $q_o = t_o - 32$ and $q_2 = t_2 - 32$ (t_o = initial and t_2 = final temperature of the water, deg. fahr.). The lb. of water delivered per lb. of steam supplied may then be expressed,

$$w = (H - t_2 + 32) \div (t_2 - t_o). \qquad (258)$$

Figure 453 gives the performance of a "Desmond" automatic injector as tested at the Armour Institute of Technology. The curves were plotted from the simplified equation (258) and the circles represent actual test data. The close agreement between calculated and observed data is evident.

Referring to Fig. 453, *A*, it will be seen that the weight of water delivered per lb. of steam decreases as the initial pressure is increased, all other factors remaining the same. From Fig. 453, *B*, it will be noted that the weight of water delivered per lb. of steam decreases as the temperature of suction supply is increased, up to a point where the injector "breaks" or becomes inoperative. This critical temperature varies with the different types of injectors, being highest for the double-tube type, but seldom exceeds 160 deg. fahr. Figure 453, *C*, shows that the weight of water delivered per lb. of steam is practically constant for all discharge pressures within the limits of the apparatus.

In selecting an injector, the following information is desirable for best results:

1. The lowest and highest steam pressure carried.
2. The temperature of the water supply.
3. The source of water supply, whether the injector is used as a lifter or non-lifter.
4. The general service, such as character of the water used, whether the injector is subject to severe jars, etc.

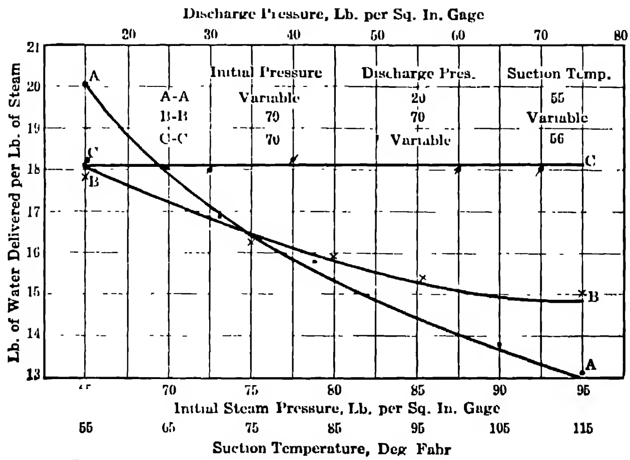


FIG. 453. Performance of an Automatic Injector.

From a purely thermodynamic standpoint, the efficiency of an injector is nearly perfect, since the heat drawn from the boiler is returned to the boiler again, less a slight radiation loss. As a pump, however, the injector is very inefficient and requires more fuel for its operation than very wasteful feed pumps. This is best illustrated by an example:

Example 72. — Compare the heat consumption of a high-grade injector with that of an ordinary duplex boiler feed pump when feeding water to a boiler. Make all necessary assumptions.

Solution. — An injector of modern design will deliver, say, 13 lb. of water per lb. of steam under the following conditions: initial steam pressure 115 lb. abs.; feedwater 60 deg. fahr.; delivery temperature 140 deg. fahr.; suction lift 3 ft. From steam tables, heat content of 1 lb. of steam at 115 lb. abs. = 1188.8 B.t.u. Neglecting radiation and friction, the heat required to deliver 1 lb. of water to the boiler =

$$[1188.8 - (140 - 60)] \div 13 = 85.3 \text{ B.t.u.}$$

A simple direct-acting duplex pump consumes, say, 200 lb. steam per i.hp-hr. Assume the extreme case where the exhaust steam will not be

used for heating the feedwater and the latter is fed into the boiler at 60 deg. fahr.

The heat supplied to the pump per i.hp-hr.,

$$200\,[1188.8 - (60 - 32)] = 232{,}160 \text{ B.t.u.}$$

Assuming the low mechanical efficiency of 50 per cent, the heat required to develop 1 hp-hr. at the water end will be

$$232{,}160 \div 0.50 = 464{,}320 \text{ B.t.u. per hr.}$$

Since the steam pressure is 100 lb. gage, the equivalent head of water at 60 deg. fahr. is

$$2.3 \times 100 = 230 \text{ ft.}$$

Assume the friction in the feed pipe, the resistance of valves, etc., to be 30 per cent of the boiler pressure; the total head pumped against will be

$$230 + 69 = 299, \text{ say } 300 \text{ ft.}$$

Since 1 hp-hr. = 1,980,000 ft. lb. per hr.,

$$\frac{1{,}980{,}000}{300} = 6600 \text{ lb. per hr.,}$$

that is, 1 hp. at the pump will deliver 6600 lb. of water per hr. to the boiler against a head of 300 ft.

The heat consumption per lb. of water delivered,

$$464{,}320/6600 = 70.3 \text{ B.t.u.}$$

Under the assumed conditions, the injector requires 85.3 B.t.u. to deliver 1 lb. of water, against 70.3 B.t.u. for the pump (with the better grades of pumps this disparity is considerably greater). This refers to the performance of the injector solely as a pumping mechanism. As a boiler feeder, however, the injector returns practically all of the 85.3 B.t.u. to the feedwater, so that its efficiency is virtually 100 per cent. Although the injector has a perfect efficiency as a boiler feeder, it is not necessarily the most economical means for feeding a boiler, because of its inability to operate with hot water, and the effect is equivalent to heating the feedwater by live steam.

277. Rotary Pumps. — Rotary pumps are occasionally used for circulating cooling water in condenser installations, and give about the same efficiency as centrifugal pumps under similar conditions of operation. For moderate pressure and large volumes, they offer the advantage of low rotative speed, thus permitting direct connection to slow-speed steam engines. At high speeds they are noisy, owing chiefly to the gearing. They occupy considerably less space than piston pumps of the same capacity, but require more room than the centrifugal type.

Figure 454 shows a section through a two-lobe cycloidal pump. The

shafts are connected by wheel gearing, the power being applied to one of the shafts. The water is drawn in at I and forced out at O, the displacement per revolution being equal to four times the volume of chamber A. There is no rubbing between impellers and casing. In this type of pump the pressure is independent of the speed of rotation, and the capacity varies almost directly with the speed. The slip varies from 5 to 20 per cent according to the discharge pressure.

Figure 455 shows a section through a rotary pump with movable butment. Figure 456 illustrates the performance of a 45-mm. Siemens-Schuckert rotary pump at different speeds and discharge pressures.

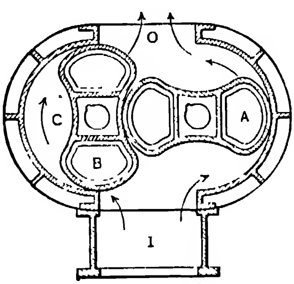


FIG. 454

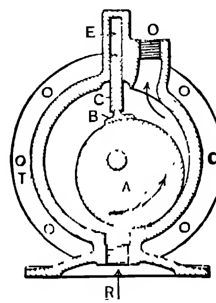


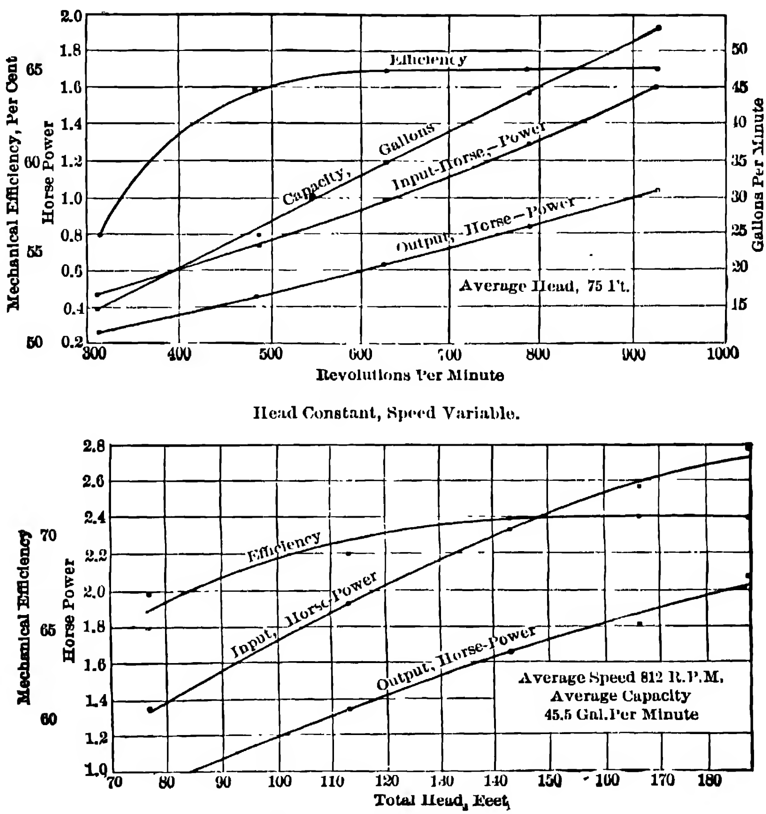
FIG. 455

(*Zeit. d. Ver. Deut. Ing.*, June 24, 1905, p. 1040.) Large rotary pumps give much higher efficiencies, but the general characteristics are about the same. A combined efficiency of pump and engine as high as 84 per cent has been recorded. (*Trans. A.S.M.E.*, Vol. 24, p. 385.)

Screw pumps may be grouped with the rotary positive-displacement class. The **Quimby** screw pump is one of the best-known examples of this type of pump and consists essentially of two right and left square-thread screws revolving in a double casing. The liquid to be pumped is drawn in at the outer ends of the cylinder and forced toward the center by the action of the two pairs of intermeshing threads. The discharge is from the center of the casing. Power is applied to one of the screws and the second is driven by means of a pair of gears. The screws run in close fit with the casing but without actual contact. Quimby pumps operate at speeds varying from 600 to 1500 r.p.m., depending upon the size and service for which they are intended.

278. Centrifugal Pumps. — There is still a wide field of application for piston pumps in small power plants and for certain industrial purposes, particularly where the quantity of fluid to be handled is small and the head pumped against is high; and, under certain conditions, the rotary or screw type of pump may be installed to advantage; but in a general

sense the centrifugal pump has practically supplanted all other types, because of its compactness, simplicity, balanced rotary motion, absence of valves and pistons, uniform pressure and flow, freedom from shock, ability to handle dirty water, and high rotative speed, permitting direct connection to electric motors or steam turbines. In the large modern power plant the boiler-feed, circulating, condensate and other auxiliary



Head Constant, Speed Variable.

Speed Constant, Head Variable.

Fig. 456

pumps are all of the centrifugal type. Efficiencies as high as 87 per cent have been realized with special designs, and 80 per cent is a common performance for the better grade of pumps, while the lift is practically limited only by the speed of the impeller. While this efficiency is not as high as that of a first-class piston pump, the other advantages more than offset this disadvantage. Triple-expansion flywheel pumping engines show higher duties and, therefore, greater heat economies than the best

turbine-driven centrifugal pumps, but that this advantage does not offset the lower first cost of the centrifugal pump equipment is evidenced by the increasing number of installations of the latter for waterworks service.

Centrifugal pumps consist of two essential elements, (1) a rotary impeller which draws in the water at its center, and (2) a stationary casing which guides the water to and from the impeller. The centrifugal force set up by rotation of the impeller throws the particles of water outward, imparting energy to them. At exit from the impeller, the gain of energy appears partly as pressure (potential energy) and partly as velocity (kinetic energy). For maximum efficiency, as much as possible of this kinetic energy must be transformed into pressure. This is accomplished in two ways, (1) by a plain casing of spiral or volute design forming a gradually increasing water or "whirlpool" chamber which minimizes shock and converts velocity head to pressure head, and (2) by fitting the casing with a series of guide or **diffusion vanes** which effect the same result. Pumps equipped with spiral casings are known as **volute** pumps, while those fitted with diffusion vanes are known as **turbine** pumps.

Figure 457 gives an end view of a typical volute pump with end plate removed so as to expose the impeller, and Fig. 458 shows a section through a modern construction. The impeller may be open as in Fig. 459, *B*, or closed as in Fig. 459, *A*. The open design is used only in the cheaper pumps and for pumping sewage. Volute pumps are usually of single-stage construction and are designed for heads of 150 ft. and under, though they are not necessarily limited to low heads and to single stages. Since the head is limited only by the peripheral speed of the impeller, it is evident that a given lift may be obtained by a large-diameter impeller revolving at low rotative speed or a small-diameter impeller operating at high rotative speed. Increase in impeller diameter, however, means increased area of frictional surface, causing a rapid increase in power loss. Therefore, the smaller the impeller diameter and the higher the rotative speed, the higher the efficiency, a condition also true of the driving member. The limiting dimension to which the diameter can be reduced is that of the inlet eye through which the water must enter at moderate velocity. For very large capacities and high speeds, several impellers operating in parallel are preferred to a single rotor, in order to keep down the size. The two-impeller design is known as a **bi-rotor pump** and the three-impeller design as a **tri-rotor pump**, and so on, depending upon the number of impellers.

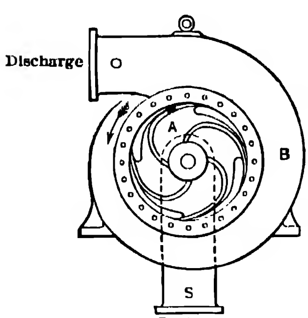


FIG. 457. Typical Centrifugal Pump.

Figure 460 shows a section through a three-stage pump illustrating the turbine type which is usually of multi-stage design. The multi-stage pump is in reality a number of single pumps arranged in series in a single

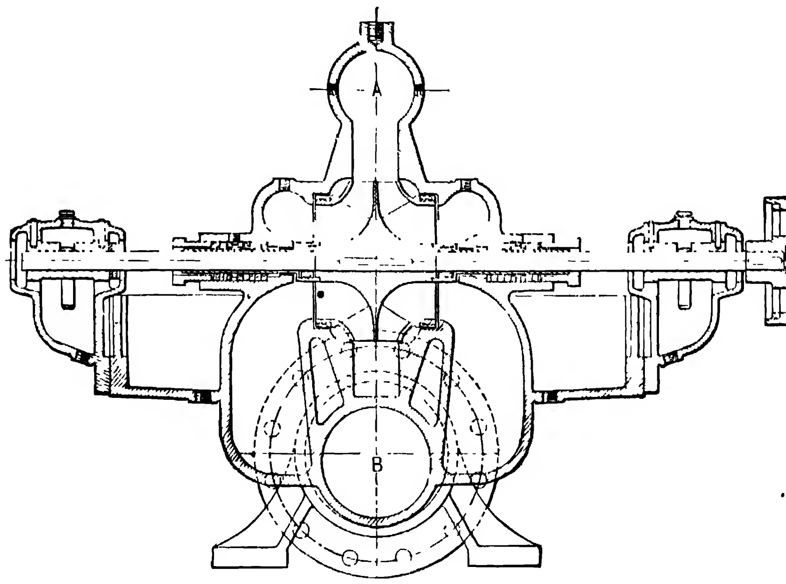


FIG. 458. Typical Single-stage, Double-suction, Volute Pump.

casing, the discharge from the first pump being directed into the suction of the second, and so on. The delivery pressure of the last stage is approximately the sum of the heads of each stage.

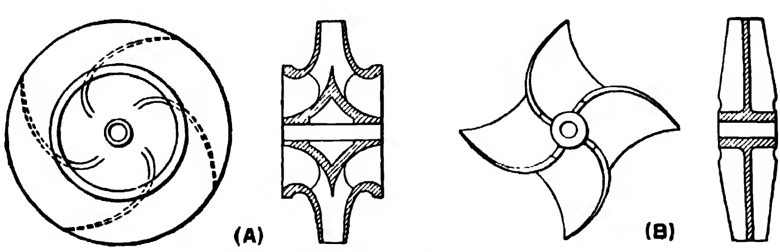


FIG. 459. Basic Types of Impellers.

Centrifugal pumps may be belted, geared, or direct-connected to any type of prime mover or auxiliary drive, the kind of drive depending upon the size, type and load characteristics of station. Being a relatively high-speed machine, it is well suited to steam-turbine and motor drives. In the older stations, practically all centrifugal pumps were steam-driven;

but in the modern plant, the tendency is toward motor drives or a combination of steam and motor drives, the distribution of steam and motor-driven auxiliaries depending upon the method of establishing the station heat balance.

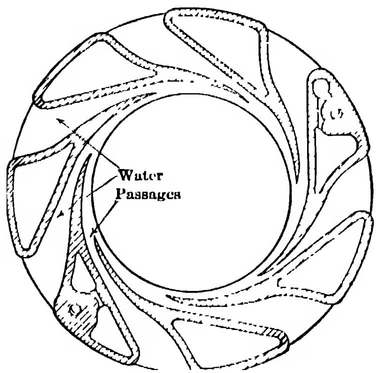


FIG. 459a. Diffusing Ring for Turbine-type Centrifugal Pump.

The average centrifugal pump requires from 20 to 30 per cent of full torque at starting and 50 to 60 per cent of full torque at approximately full speed, provided the discharge valve is closed. If the discharge valve is open during the starting period the initial torque will be the same, but full-load torque will be required at normal speed. With direct-current supply the motor drive for constant speed may be either of the shunt-wound or compound series. The discharge valve of the pump should be closed in starting shunt-wound motors and preferably so with the compound. For variable speed the compound motor with about 10 per cent series winding is ordinarily employed, whether speed adjustment is by field or armature operation.

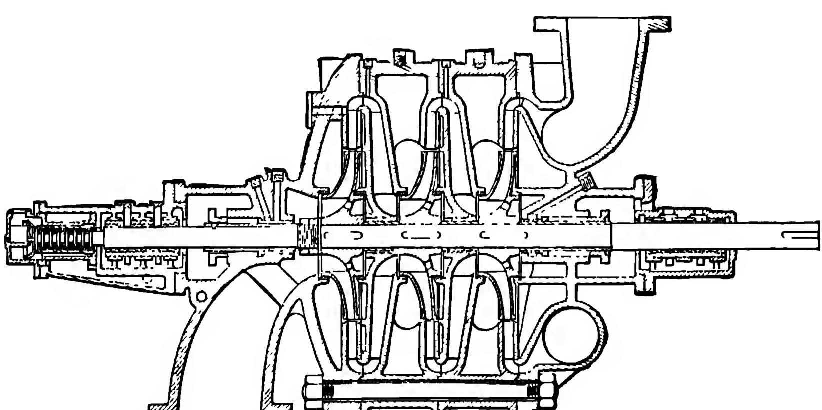


FIG. 460. Worthington Three-stage Turbine Pump.

With alternating-current supply, the synchronous, squirrel-cage, slip-ring and brush-shifting types of motors are used, depending upon the nature of the service and the electric system available. Synchronous motors are applicable only to the larger pumps operating uninterruptedly at constant speed against constant head, but are favored because of the

possible power-factor correction and first cost. The application of this type of motor to centrifugal pumps is restricted to some extent by its comparatively low speed limitation. The squirrel-cage motor has a definite limit to the starting torque it will develop and cannot be started by a direct connection across the line except in small sizes. The general method of starting is to apply reduced voltage to the primary member and, when the rotor is up to speed, to throw it on the full-line voltage. For adjustable-speed operation, the slip-ring and the brush-shifting types are used. The latter is recommended where considerable speed reduction is to be effected. The curves in Fig. 461 show the relation between input and capacity for a centrifugal pump driven by different types of motors, and serve also to show the desirability of varying pump capacity by change of speed instead of by throttling. For maximum efficiency and satisfactory operation, the pump and drive should be considered as a unit and selected accordingly.

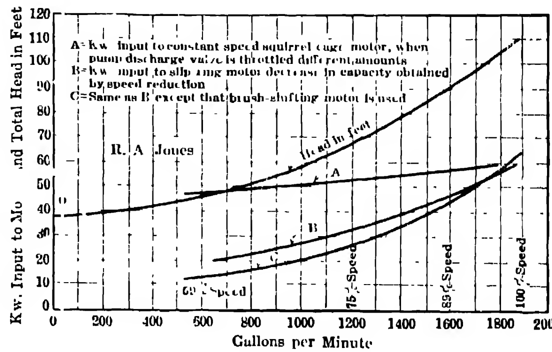


FIG. 461. Relation between Input and Capacity for Different Types of Motors.

Centrifugal Pumps: R. L. Daugherty, McGraw-Hill Book Co.

The Centrifugal Pump: Power Plant Engrg., Feb. 15, 1921, p. 218; Aug. 15, 1920, p. 785; Power, May 3, 1921, p. 698; May 17, 1921, p. 779.

Fitting the Electric Motor to the Pump: Power, Dec. 18, 1923, p. 976.

Electric-motor Drives for Pumps: Power, July 5, 1921, p. 2.

Induction Motors Driving Centrifugal Pumps: Power, Aug. 26, 1919, p. 324.

Driving Power-house Auxiliaries: Power, Jan. 31, 1922, p. 166.

Electric Motors for Driving Pumps: Power, Sept. 5, 1922, p. 363.

279. Performance of Centrifugal Pumps. — The design and theory of centrifugal pumps is beyond the scope of this text, and the reader is referred to the accompanying bibliography for extended study. The fundamental principles involved in the performance of centrifugal pumps are similar to those of centrifugal fans and may be briefly stated as follows:

(1) The speed, V, in ft. per sec., of a point on the periphery of the impeller is equal in velocity to water falling from the same height as the head in ft. pumped against, h, or $V = \sqrt{2gh}$, in which g = acceleration of gravity or 32.2 ft. per sec. Conversely, the maximum theoretical head or lift is $h = V^2/2g$.

(2) For a constant diameter of impeller (*a*) the quantity pumped will vary as the speed, (*b*) the head will vary as the square of the speed, and (*c*) the power will vary as the cube of the speed.

(3) For a constant speed and change in diameter of the impeller, (*a*) the quantity pumped varies as the diameter of the impeller, (*b*) the head varies as the square of the diameter, and (*c*) the power varies as the cube of the diameter.

These laws are not strictly true, but the departure is small.

Example 73. — The impeller of a centrifugal pump is 15 in. in diameter. At what speed must it operate to lift water to a height of 100 ft?

Solution. — $V = \sqrt{2\,gh} = \sqrt{64.4 \times 100} = 80.4$ ft. per sec. or 4824 ft. per min.

$$V = 2\,\pi r n$$

$$4824 = 6.28 \times 0.625 \times n. \quad n = 1230 \text{ r.p.m.}$$

This is the speed necessary to lift the water to a height of 100 ft., but in order to actually deliver water the speed must be increased in order to overcome friction and impart velocity to the water.

The velocity of water at the discharge opening of the pump varies in practice from 5 to 15 ft. per sec. A good working range is 10 to 12 ft. per sec. The head corresponding to the velocity of discharge may be obtained by substituting the discharge velocity in ft. per sec. for V in the preceding equation and solving for h. This quantity is ordinarily so small that it may be neglected. The friction head may be estimated as shown in paragraph 309.

The suitability of a centrifugal pump for a given service is determined from **characteristic curves** showing the relation of head, speed, capacity, power and efficiency. These curves are based on actual test results and vary with the design of pump. The relationship between the various quantities is largely controlled by the angles and curvatures of the impeller blades, and the shape of the volute, or arrangement and design of the diffusion vanes. If the vanes are radial or inclined forward in the direction of rotation, the head will increase with increased delivery, while if they are curved backwards sufficiently, the head will remain constant or fall off as the delivery decreases. For each set of operating conditions, there are certain characteristics which give the best results, and it is the endeavor of the manufacturer to design his pumps to meet these requirements. The usual form of characteristic curves is based on constant speed, the curves showing the relation between head, capacity, efficiency, and brake horsepower at this speed. Many other curves can be obtained, however, by keeping any one of the fundamental quantities constant, and

by varying the others. Ordinates and abscissas are ordinarily expressed directly in the quantities as observed and calculated (see Fig. 462), but quite frequently they are based on percentages, as in Fig. 463. The interpretation of these curves is the same as for fan characteristics (see

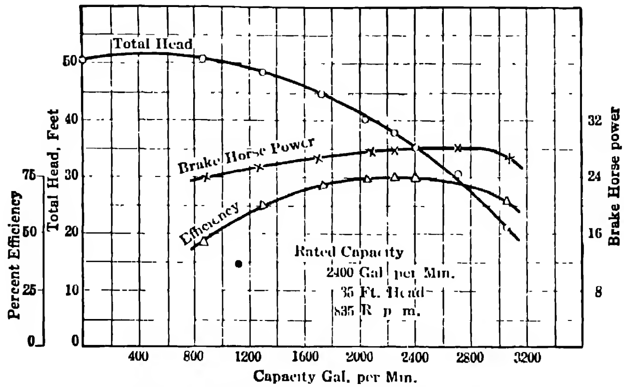


FIG. 462. Test Characteristics of a 10-in. Wheeler Centrifugal Pump.

paragraph 158) and need not be discussed here. Since manufacturers furnish curves for their specific product, and the performances vary within wide limits, general curves are without purpose except for rough approximations.

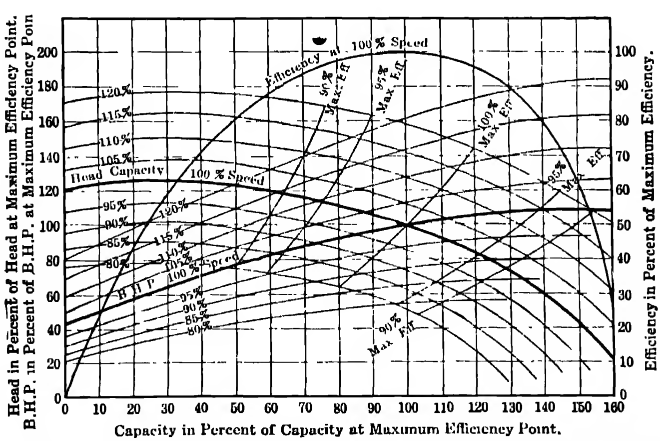


FIG. 463. Characteristics of a Worthington "Type-VH Class-B" Centrifugal Pump.

It has been stated that, for a given pump, the quantity pumped varies directly with the speed, the head with the square of the speed, and the power with the cube of the speed. The following examples illustrate the application of these laws to a specific case.

Example 74. — Using the data in Fig. 462, calculate the capacity, heads, power and efficiency, if the speed is increased from 835 to 1000 r.p.m.

Solution. — Rated capacity at 900 r.p.m. is 2400 gal. per min. Capacity at 1000 r.p.m. = 2400 × 1000 ÷ 835 = 2874 gal. per min.

Head at 900 r.p.m. and rated capacity is 35 ft. Head at 1000 r.p.m. = 35 × $(1000 \div 835)^2$ = 50.2 ft.

B.hp. at 900 r.p.m. and rated capacity is 35. Br.hp. at 1000 r.p.m. = 28 × $(1000 \div 835)^3$ = 48. (Actual tests of the pump in question at 1000 r.p.m. gave the following results: Capacity, 2850 gal. per min.; head, 51.5 ft.; and power, 46.5 br.hp.)

If the laws just cited are strictly true, the efficiency at 1000 r.p.m. must necessarily be the same as at 835 r.p.m., since the product (1000 ÷ 835) $(1000 \div 835)^2$ in the numerator is cancelled by $(1000 \div 835)^3$ in the denominator, thus:

$$\text{Eff.} = \frac{\text{Total head (ft.)} \times \text{capacity (lb. per hr.)}}{33{,}000 \times \text{br.hp.}}$$

$$\text{Eff. at 835 r.p.m.} = \frac{35 \times 2400 \times 8.35}{33{,}000 \times 28} = 0.76.$$

$$\text{Eff. at 1000 r.p.m.} = \frac{35\,\frac{1000}{835} \times 2400\left(\frac{1000}{835}\right)^2 \times 8.35}{33{,}000 \times 28\left(\frac{1000}{835}\right)^3} = 0.76.$$

(Actual test efficiency = 0.798.)

Size does not influence the efficiency of a centrifugal pump, provided the combination of head, capacity, and speed is favorable; but for the conditions usually met with in practice, the following efficiencies are conservative for rough approximations.

Normal Capacity Gal. per Min.	Eff Per Cent		Normal Capacity Gal per Min.	Eff. Per Cent	
	A	B		A	B
100– 150	50	45	1500–1800	72	70
200– 350	55	50	2000–3000	75	72
400– 600	60	56	3500–4500	76	73
650– 900	65	62	5000–6500	77	74
950–1300	70	68	Over 6500	78	75

A. Single-stage up to 150 ft head. *B.* Multi-stage over 150 ft. head.

Efficiencies as high as 87 per cent have been realized with special designs when operating under favorable conditions, and 80 per cent is common practice with the larger and better grades of modern pumps, so that the values given above should be considered as " average " only.

Characteristic Curves of Centrifugal Pumps. Power, Oct. 23, 1923, p. 653.
Centrifugal Pumps: Power, Mar. 3, 1921, p. 698; May 17, 1921, p. 780; May 20, 1919, p. 763.
Centrifugal Pumps: Power Plant Engrg., Aug. 15, 1920, p 785; Feb. 15, 1921, p. 218.
Parallel Discharge of Centrifugal Pumps Power, Aug. 6, 1920, p. 554.

280. Vacuum Pumps. — The different types of vacuum pumps employed in steam power plant practice may be divided into four general classes:

1. Wet-air pumps.
2. Tail or removal Pumps.
3. Dry-air pumps.
4. Condensate pumps.

(1) *Wet-air pumps* are for the purpose of withdrawing water and non-condensable gases from apparatus under less than atmospheric pressure. Standard low-level jet-condenser wet-air pumps handle simultaneously the circulating water, condensate, and all entrained air and are, in fact,

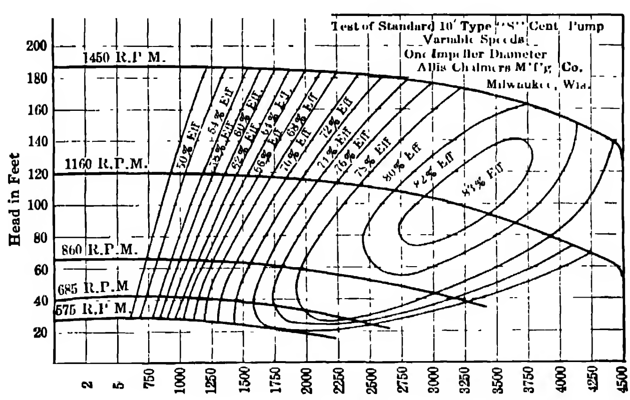


FIG. 464. Characteristics of an Allis-Chalmers "Type-S" Centrifugal Pump.

a combination of circulating pump and vacuum pump. Surface-condenser wet-air pumps deal with the condensate and its air entrainment. Wet-air pumps may be of the reciprocating, centrifugal, rotary-jet, rotary, positive-displacement, or steam-jet type.

(2) The terms "wet-vacuum pump," "wet-air pump," and "tail pump" are often used synonymously, but in order to differentiate between pumps handling injection water, condensate, and air, and those dealing only with the injection water and condensate, the term "wet-air pump" has been applied to the former and "tail pump" to the latter.

(3) *Dry-air pumps* are for the purpose of withdrawing the non-condensable gas and entrained vapor from apparatus under a vacuum and discharging it against atmospheric or greater pressure. They are, to all intents and purposes, air compressors. The term " dry air " is a misnomer since the gases exhausted are almost invariably saturated with water vapor. These pumps may be of the reciprocating, rotary, positive-displacement, hydro-centrifugal, or steam-jet types.

(4) *Condensate pumps* are for the purpose of withdrawing condensed steam from surface condensers and are usually of the reciprocating, rotative or centrifugal types.

281. Wet-air Pumps for Jet Condensers. — Figure 465 shows a section of the cylinder of a Dean twin-cylinder wet-air pump as applied to a standard low-level jet condenser and is illustrative of the reciprocating type. There are three sets of valves, the suction or foot valves A, A, the lifting or bucket valves B, B, and the head or discharge valves C, C. On the upward stroke of the piston or bucket, a partial vacuum is formed in the chamber between the bucket and the lower head, causing the water and air in the bottom of the barrel to lift the foot valves A, A from their seats and flow into the cylinder. On the downward stroke, the foot valves A, A close and water and air are entrapped in chamber R between the lower head and the bucket. As the bucket descends, the pressure of air in the cylinder lifts the bucket valves B, B from their seats and permits the air and water to escape to the upper portion S of the cylinder between the head plate and the bucket. On the next upward stroke, the water and air are forced through the discharge valves C, C into the hotwell. This discharge of water and air from the top compartment is simultaneous with influx of water and air in the lower chamber.

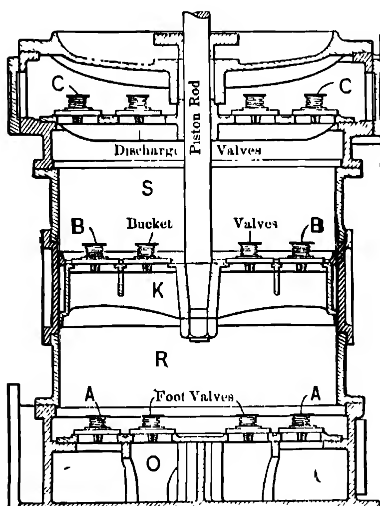


Fig. 465. Dean Wet-air Pump.

Figure 466 shows a vertical section and sectional end elevation of a **Rees Roturbo** rotary-jet condenser illustrating an adaptation of the rotary-jet pump as a jet condenser. This pump is a development of a special type of centrifugal pump, the unique feature of which is the employment of a revolving pressure chamber. The hollow impeller, Fig.

466, lifts the circulating water in much the same manner as in any centrifugal pump. The space between the periphery of the impeller and the

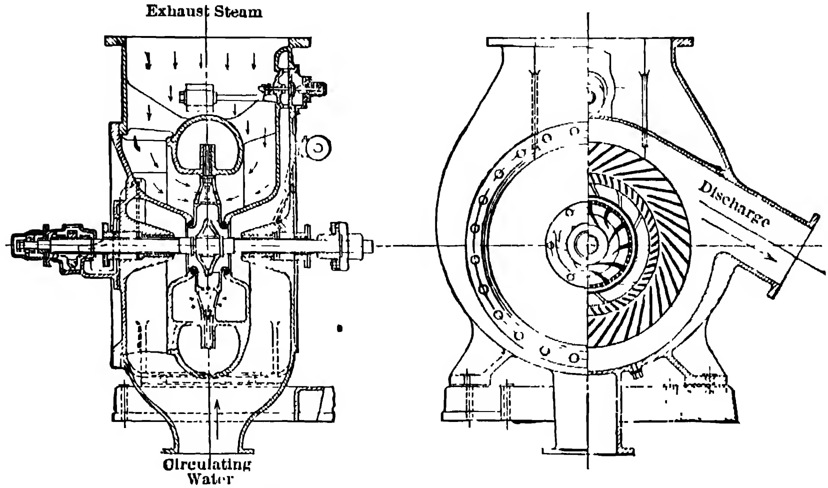


FIG. 466. Rees "Roturbo" Jet Condenser.

inner circumference of the fan wheel forms the mixing chamber in which the exhaust steam is brought into contact with radial jets of water. The fan wheel itself acts as an ejector and exhausts the mixture of circulating water and vapor. The operation is as follows: circulating water is drawn through the suction pipe into the revolving pressure chamber, on the periphery of which nozzles are arranged as shown in Fig. 467, and is forced through the nozzles in radiating jets which are arranged to impinge in pairs. The water jets, which are made fan-shaped and subdivided into a fine spray, are projected in lines radiating from the shaft (but still rotating as a whole with the impeller) across a space into which the exhaust steam blows. The circulating water leaving the nozzles, condensate, and air entrainment are picked up by the blades of the fan and discharged through a volute guide chamber to the hotwell.

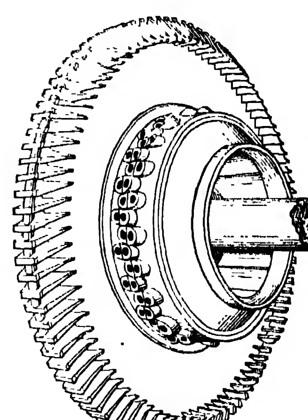


FIG. 467. Impeller for Rees "Roturbo" Jet Condenser Pump.

The Connersville jet condenser is a typical example of an application of a rotary positive-displacement wet-air pump. In this device the cir-

culating water, condensate, and air entrainment are handled by a **Connersville**, cycloidal, 3-lobe type rotary pump. (A cross section through a typical 2-lobe cycloidal pump is shown in Fig. 454.)

The steam-jet type of wet-air pump is exemplified in the ejector condenser. See paragraph 285.

282. Wet-air Pumps for Surface Condensers. — These pumps exhaust the condensate and air entrainment from surface condensers. The vacuum pumps of a steam heating system also come under this head.

The **Edwards air pump,** Fig. 468, is a typical example of a wet-air pump of the reciprocating type. Referring to Fig. 468, the condensed steam flows continuously by gravity from the condenser into the base of the pump through passage A and annular space B. As the piston C descends, it forces the water from the lower part of the casing F into the cylinder proper through the ports P, P. On the upward stroke the ports in the piston are closed and the air and water discharged through head valves D and exhaust port E to the hotwell. The seats of valves D are constructed with a rib between each valve and a lip around the outer edge, so that each valve is water-sealed independently of the others. In ordinary air pumps, the clearance between the bucket and head-valve seat is necessarily large, due to the space occupied by the bucket valves and the ribs on the under side of the valve seating. This clearance space reduces the capacity of the pump, since the air above the bucket must be compressed above atmospheric pressure before it can be discharged, and on the return stroke will expand and occupy a space which should be available for a fresh supply of air from the condenser. In the Edwards air pump the clearance space is reduced to a minimum, since there are no bucket valves to limit it. The absence of suction or foot valves still further increases the capacity of the pump for similar reasons. These pumps are arranged either single, double, or triplex; steam, electric, or belt-driven; slow or high speed.

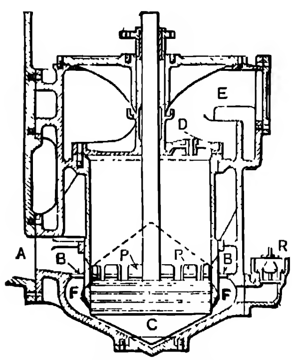


FIG. 468. Edwards Air Pump.

Figure 469 shows a partial axial and an end section through a **C. H. Wheeler** Manufacturing Co. high-vacuum **"Rotrex"** pump. This pump is of the wet-vacuum type and handles both air and water of condensation but it is also adapted for dry-air purposes. The apparatus consists of a cylindrical casing and a rotor mounted eccentrically on the shaft. This shaft is carried in outboard ring oil bearings which are entirely independent of the stuffing boxes. The division between the suction and discharge

space in the pump cylinder is maintained by a radius cam carried on a shaft independent of the stuffing boxes. This cam is operated from the rotor shaft by a lever and crank on the outside of the casing. The clearance spaces are water-sealed. The discharge valves are of the Gutermuth type. Pump speed 200 to 300 r.p.m. The manufacturers guarantee

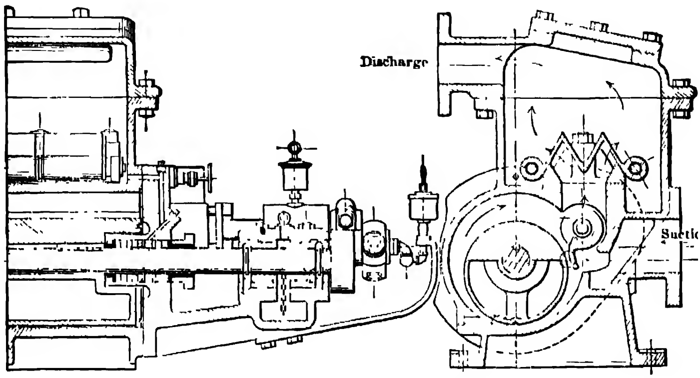


FIG. 469. High-vacuum Rotrex Pump

that on dead-end test a vacuum may be obtained within one-half inch of the barometer, and within one inch of the barometer under operating conditions.

283. Size of Wet-air Pumps. — Since the wet-air pump for a jet condenser must deal with the mixture of injection water, condensate, and all air entrainment, the problem of design is essentially that of determining the volume of mixture to be withdrawn under condenser pressures and temperatures. The volume of injection water and condensate for a given set of conditions may be readily calculated, but the volume of air entrained with the injection water and condensate and that introduced by leakage is an unknown quantity and can only be estimated. The amount of air mechanically mixed with the injection may vary from 1 to 5 per cent by volume at atmospheric pressure and temperature. The amount of air in feedwater varies from less than 1 per cent by volume, if the heater is of the open type, to 5 per cent or more if the heater is of the closed type and raw water is fed directly into the heater. Air leakage is an unknown quantity varying within wide limits, and is dependent upon the tightness of joints, stuffing boxes and the like. A very liberal factor is usually allowed for air entrainment, leakage, and pump slip, an average figure being about 10 per cent by volume of the circulating water for the combined air and wet-vacuum pump for jet condensers and 10 per cent by volume of the feedwater for surface condensers.

Let Q = total volume of air and water, in cu. ft. per hr., to be handled by the pump,
V = volume of cooling water in cu. ft. per hr.,
v = volume of condensed steam in cu. ft. per hr.,
v_a = volume of air-vapor mixture to be exhausted per hr.,
w_1 = air leakage, lb. per hr.,
t_a = temperature of the air in the condenser, deg. fahr.,
t_2 = temperature of the discharge water, deg. fahr.,
t_0 = initial temperature of the cooling water, deg. fahr.,
p_a = atmospheric pressure, in. of mercury,
p_c = total pressure in the condenser, in. of mercury,
p_v = pressure of aqueous vapor at temperature t_2,
then $(V + v)$ = volume of water to be pumped from the condenser per hr.

The volume of air-vapor mixture to be removed per hr. may be calculated from equation (135), thus

$$v_a = w_1 \frac{0.754\,T_a}{p_c - p_v} \tag{259}$$

And the total volume to be exhausted per hr. by the pump is

$$Q = V + v + w_1 \frac{0.754\,T_a}{p_c - p_v}. \tag{260}$$

Example 76. — Calculate the piston displacement of a wet-air pump suitable for a 1000-hp. piston-engine plant operating under the following conditions: Water rate 16 lb. per i.hp-hr.; initial steam pressure 150 lb. abs.; vacuum, 4 in. abs.; injection water 70 deg. fahr.; hotwell 110 deg. fahr.; air leakage and entrainment 7.5 lb. per thousand cu. ft. of injection water.

Solution. — Here $p_c = 4$, p_v (from steam tables) $= 2.59$, $t_0 = 70$, $t_2 = t_a = 110$, $v = 0.04\,V$ (from equation 204), $w_1 = 0.0075\,V$ (by assumption). Substituting these values in equation (260) and solving

$$Q = V + 0.04\,V + 0.0075\,V \frac{0.754 \times 570}{4.00 - 2.59} = 3.33\,V.$$

Average practice gives 3 V as the piston displacement per hr. for a single-acting pump and 3.5 V for a double-acting pump, the cylinders being ordinarily proportioned on a piston velocity of 50 ft. per min. at rated capacity.

Wet-air pumps are usually independently driven, making it possible to vary the speed of the pump irrespective of the engine speed and to create a vacuum before starting the engine. Occasionally, however, when the load is constant, as in pumping-engine practice, the pump may be driven by the main engine.

The combined air, condensate and circulating pump (with the exception of pumps of the Rees " roturbo jet " type) is not adapted for high-vacuum work on account of the enormous increase in air volume at very low pressures. With cold injection water and a good air-tight condensing system, vacua as high as 2 in. abs. are possible with the standard type of jet-condenser air pumps, but practice recommends the use of separate air and wet-vacuum pumps for vacua higher than 26 in.

Since the wet-air pump for surface condenser handles only the condensed steam and air, its theoretical capacity, neglecting clearance, may be determined by eliminating V from equation (260), which then becomes

$$Q = v + w_1 \frac{0.754 T_a}{p_c - p_v}. \tag{261}$$

The volume of air entering the condenser varies so much with the character of the power plant equipment and the conditions of operation that any assumed average value of v_a may lead to serious error.

Average steam-turbine practice gives

$Q = 20\ v$ for 26-in. vacuum,
$Q = 30\ v$ for 27-in. vacuum,
$Q = 40\ v$ for 28-in. vacuum,
$Q = 50\ v$ for 29-in. vacuum.

Average reciprocating engine practice gives

$Q = 85$ per cent of above for vacua up to 27 in.

284. Tail or Removal Pumps. — As previously stated, the term " tail " pump has been applied to pumps which deal with the combined circulating water and condensate, merely to distinguish between this type and that dealing with the entire condenser-water supply including the air entrainment. In practice the terms tail pump and wet-air pump are used synonymously. Almost any type of water pump may be used for the purpose of withdrawing the combined circulating water and condensates, but the centrifugal pump appears to be the more common in use. A typical tail-pump installation is shown in Fig. 370. The Leblanc jet condenser, Fig. 339, and the C. H. Wheeler low-head high-vacuum jet condenser,

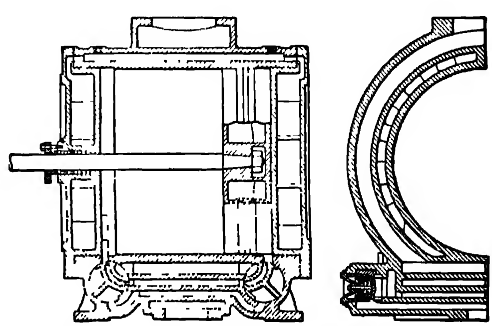


Fig. 470. Air Cylinder Construction of Wheeler Dry-Vacuum Pump.

Fig. 340, involve the use of centrifugal tail pumps. The power required to drive this style of pump may be calculated from equation (262). In this connection the total head pumped against must include the suction head due to the vacuum in the condenser.

285. Dry-air or Dry-vacuum Pumps. — Dry-air or dry-vacuum pumps are used in connection with jet or surface condensers where a high degree of vacuum is essential, as in steam-turbine practice. Such pumps are intended to exhaust the saturated non-condensable vapors only, and are in reality air compressors. Air pumps for jet condensers must deal with much larger volumes of air than those for surface condensers, other things being equal, because of the air entrained with the circulating water. Dry-air pumps may be divided into four general groups: (1) reciprocating-piston, (2) positive rotary-displacement, (3) hydro-centrifugal and (4) steam-jet.

Piston Type: Figure 470 shows a section through the cylinder of a Wheeler dry-vacuum pump, illustrating the single-cylinder, single-stage reciprocating-piston group. The admission valves *A* and *A* are mechanically controlled and the discharge valves are of the usual spring-loaded type. The rotary admission valves are adjusted so that for a short instant at dead center communication is established between both ends of the cylinder so as to reduce the air pressure in the clearance space down to the suction pressure on the other side of the piston.

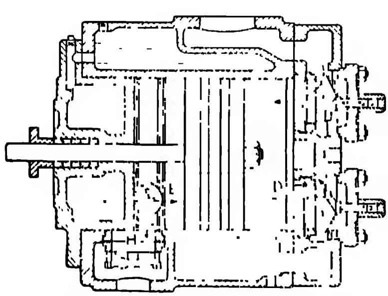


FIG. 471. Two-stage Single-cylinder Dry-air Pump.

Figure 471 shows a section through the cylinder of a **Laidlaw Feather-Valve** single-cylinder two-stage dry-vacuum pump which possesses some advantages over the single-cylinder mechanism in that a two-stage effect is produced in one cylinder. The pump is single acting, but the higher volumetric efficiency practically balances the double-acting feature in the ordinary single-stage pump and permits the use of practically the same size pump for a given capacity. The valves are thin strips of ribbon steel similar in appearance to clock-spring stock. These flexible strips seat tightly on ground-faced slotted seats, and in opening flex against solid curved guards, the ends remaining in contact with the seat at all times. Mechanically actuated valves are entirely absent. The cycle of operation is as follows: With piston moving as indicated, air is drawn into the head-end of the cylinder until the piston reaches the end of its stroke. On the return stroke, the air drawn in the head end of the cylinder is trans-

ferred (at condenser pressure) through passage D and valve E to the crank end of the cylinder. On the next stroke, the air charge is compressed through valve H to somewhat more than atmospheric pressure.

Hydraulic Type: The **Leblanc Air Pump,** Fig. 339, **Wheeler Turbo-air Pump,** Fig. 472, and the **Worthington Hydraulic Vacuum** Pump, Fig. 473, are well-known examples of the hydraulic or hurling-water dry-air pumps. They differ very little from each other in principle but vary widely in mechanical construction. In these pumps, entraining or hurling water is taken from a circulating tank and hurled by centrifugal force in thin sheets or "pistons" into a diffuser or discharge cone, each sheet or piston carrying with it a layer of saturated air drawn in from the condenser. The water is used over and over again with an addition of about 10 per cent makeup to keep down the temperature, since very little heat is abstracted from the air-vapor mixture. In some installations, the hurling water is recirculated through cooling coils so that discharge to overflow and makeup for lowering the temperature are not necessary. The cooling water used is by-passed around the condenser, being taken from the discharge of the circulating pump and returned into the discharge pipe immediately after the condenser. The hydraulic type of air pump is used in large condenser installations in preference to the reciprocating-piston type chiefly because of its compactness, high air-removal capacity, and ability to carry overloads. The reciprocating pump shows a decreasing capacity with increase in vacuum and finally reaches a point where the capacity becomes zero. Owing to the increased water velocity at high vacua, the hydraulic air pump increases its capacity as the vacuum increases. The hydraulic air pump, however, requires from two to three times as much power as the piston pump and is slow in starting. These pumps are invariably of the high-speed type and are driven by steam turbines or motors.

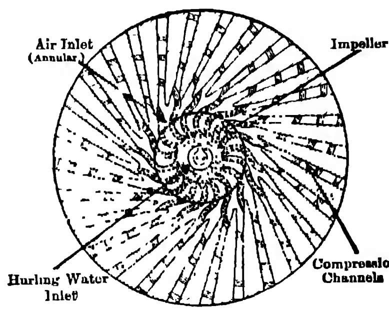


FIG. 472. Diagrammatic Arrangement of Elements in a Wheeler Turbo-air Pump.

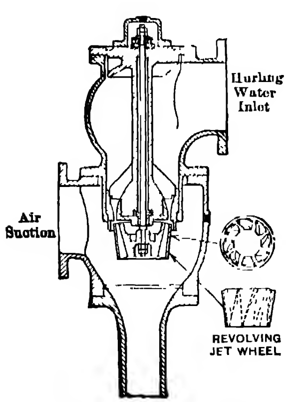


Fig. 473. Worthington Hydraulic Vacuum Pump.

Steam-Jet Type: The modern steam-jet air pump has practically sup-

planted other types for steam condensers, because of its compactness, total absence of moving parts, simplicity of operation, and high efficiency. The **Parsons vacuum augmentor,** Fig. 474, was one of the earliest practical applications of a steam jet to condenser operation. In this connection, the jet merely acted as a booster and increased the air pressure by a few inches, so that, with a vacuum of 28 in. in the condenser, the vacuum at the air pump suction would be about 26 in.

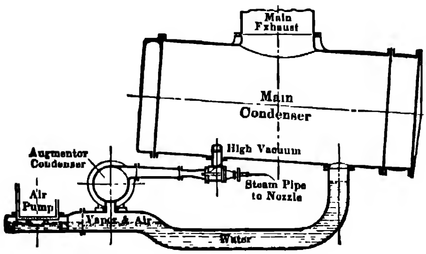


FIG. 474. Parsons Vacuum Augmentor.

The modern steam-jet or **air-ejector** is constructed in a variety of forms, but for the purpose at hand may be conveniently classified as (1) **single-stage** in which the compression is effected in one set of nozzles, and (2) **two-stage** in which the first stage discharges into the suction opening of a secondary stage. Both the single and two-stage machines may be operated condensing or non-condensing. If the cooling takes place between the first and second stage, the design is designated as of the **inter-cooler** type, and if the second stage is also equipped with a cooler, the design is designated as of the **inter-after-cooler** type. The nozzles are of either the **single** or **multi-jet** type.

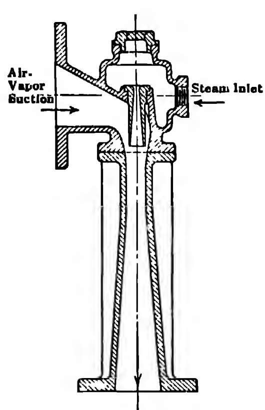


FIG. 475. Typical Single-stage Single-nozzle Ejector.

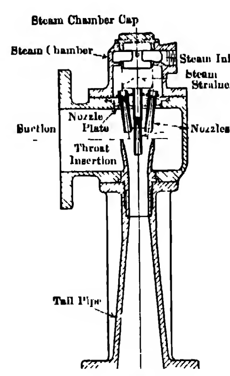


FIG. 476. Single-stage Multi-nozzle Ejector.

Figure 475 shows a section through a typical single-stage single-nozzle ejector consisting essentially of a single divergent steam nozzle discharging into the conventional form of compression tube. Steam issues from the nozzle at a velocity of 2000 to 4000 ft. per sec., depending upon the

initial steam conditions and back pressure, draws in the air-vapor mixture around the nozzle and discharges it through the compression tube against a greater pressure than that existing at the suction. By employing a number of nozzles in place of the single nozzle, Fig. 476, the air entrainment capacity may be greatly increased for the same weight of steam discharged. Careful experimental work has shown that the maximum economical compression ratio in a single ejector should not exceed about eight to one. While it is possible to obtain a vacuum within one inch of mercury abs. by a single-stage ejector, experiment shows that the steam consumption is very high for compression ratios exceeding one to eight. The single-stage machine is suitable for installations in which the discharge from the nozzle can be utilized for feedwater heating or where vacua higher than three in. abs. are not essential.

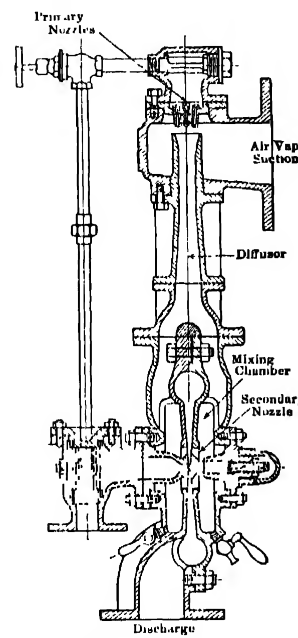


FIG. 477. Radojet Pump without Inter-cooler.

The **C. H. Wheeler radojet pump** was one of the earliest American designs involving the compound or two-stage principle and was largely instrumental in popularizing the ejector type of pump for condenser service. In this design, the primary jet withdraws the saturated air from the condenser and compresses it to four or five in. above condenser pressure and the secondary jet picks up the discharge from the primary and forces it out against the existing back pressure. The secondary jet is radial in form and discharges into an annular volute chamber. This form of nozzle causes the steam to spread out into a disc shape and in a direction which is perpendicular to the axis of the steam nozzle. This permits of an enlargement of the entrainment surface for a given mass of steam and also allows both sides of this disc-like jet to entrain the air in passing across the second-stage suction chamber.

By placing an inter-cooler, either of the jet or surface type, between the two stages, the steam from the first-stage jets is entirely condensed. The second stage, therefore, has only air to compress and since the air is but a small portion of the air-vapor mixture from the primary stage, the steam consumption of the second stage is greatly reduced and the total steam consumption of the combined stages is one-half that of a single-stage machine of equivalent air capacity.

Figure 478 shows a section through a Westinghouse steam-jet air pump of the jet inter-condenser type illustrating a modern application of the steam ejector for air-removal purposes. Steam enters as indicated and is led to the primary and secondary stage nozzles through suitable strainers. Leaving the primary nozzles at high velocity, the steam and entrained air are delivered through the primary compression tube to the inter-cooler at a pressure of four or five in. of mercury. The steam is condensed in the inter-cooler by contact with cooling water. The latter is taken from the circulating water supply if the main condenser is of the jet type or from the hotwell if the condenser is of the surface type. Air entrainment passes from the primary discharge to the secondary suction chamber. From this point the air is discharged with the steam from the secondary nozzles through the secondary compression into the atmosphere, while the water from the inter-cooler together with the condensed steam from the primary stage is drawn back to the main condenser hotwell through a looped pipe. The back pressure at the discharge opening of the secondary stage should not exceed 1 lb. gage, otherwise considerable increase in initial steam pressure would be necessary and the steam consumption would be greatly increased. Without the inter-cooler, the same quantity of air-vapor mixture can be handled, but a much larger quantity of steam would be necessary for operating the second stage.

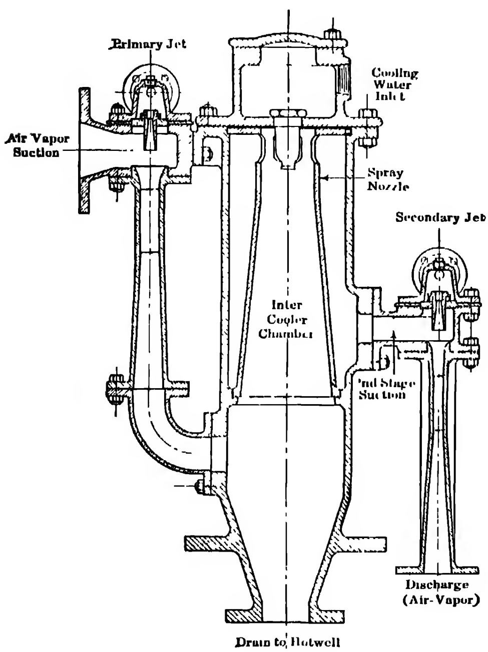


FIG. 478. Westinghouse Steam-jet Air Pump (Jet Inter-cooler Type).

Figure 479 shows a section through a C. H. Wheeler " Radojet " air pump of the **inter-after-cooler** type illustrating the latest practice in this class of pump. This design is similar to the surface inter-condenser type, with the addition of a compartment for condensing the steam from the secondary ejector. As this " after-condenser " is of the surface type, the heat from the secondary steam is absorbed by the water flowing through

the tubes, while the air escapes through the vent as shown in the illustration. There is no mixing of air and water. The passages are arranged so that the water flows first through the inter-condenser and then through the after-condenser.

With air pumps of the inter-after-cooler type, no other air-removal equipment is necessary.

286. Size of Dry-air Pumps. — The volumetric capacity of a dry-air pump for condenser service is based upon experience rather than theory, because the amount of air in the steam and the air filtration are very uncertain quantities. Since the air to be dealt with is saturated with water vapor, the pump displacement or its equivalent will be much larger than if dry air only were supplied. The weight of water vapor which must be exhausted for a given weight of dry air for different vacua and air-pump suction temperatures is shown in Fig. 335. The great reduction in volume effected by cooling the air-pump suction is clearly shown. The marked superiority of counter-current over parallel-current flow in the older designs of jet condensers is chiefly due to the greater reduction in temperature of the air and its vapor content.

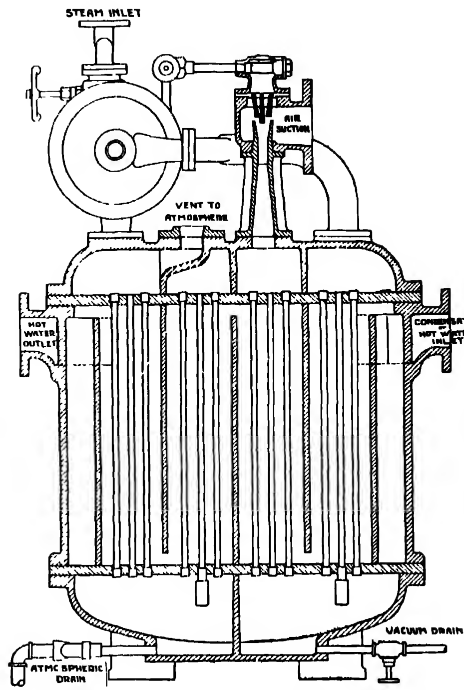


FIG. 479. Radojet Air Pump with Inter-after-cooler.

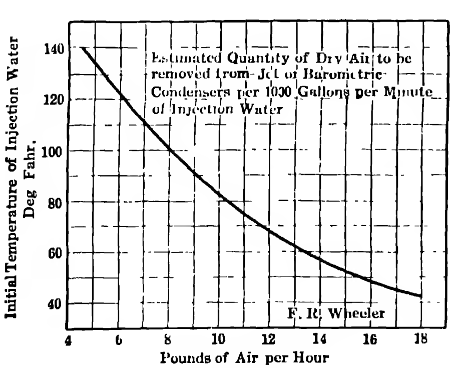


FIG. 480.

The curves in Figs. 480 and 481 may be used as a guide in estimating the weight of dry air to be handled by a dry-air pump under different vacua and tem-

peratures, but they must be used with caution since they do not allow for excessive air leakage. These curves give the weight of dry air only. In order, however, to exhaust the given quantity of dry air, the vapor entrainment must also be withdrawn. The ratio of vapor to dry air in the saturated mixture may be calculated from equation (195) or it may be taken directly from the curves in Fig. 335. The applications of these curves are best illustrated by a specific example.

Example 77. — Required the air pump capacity for a 10,000-kw. surface condenser installation using 125,000 lb. of steam per hr., vacuum 28.5 in., inlet and outlet temperature of the circulating water 70 and 80 deg. fahr. respectively.

Solution. — From Fig. 481 the dry air leakage corresponding to 125,000 lb. per hr. is found to be 33 lb. per hr. Assuming that the air-vapor mixture is withdrawn at a temperature corresponding to the mean of

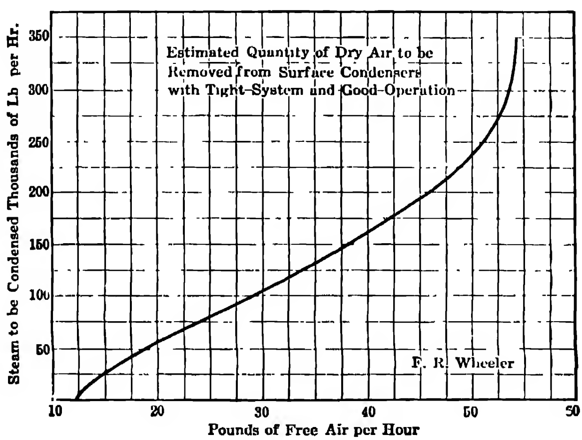


FIG. 481.

the circulating water (= 75 deg.), we find from Fig. 335 that the ratio of water vapor to dry air at this temperature and absolute pressure of 1.5 in. is 0.89. Therefore, the air pump capacity is $33 \times 1.89 = 62 +$ lb. of air-vapor mixture per hr.

It is usual, for surface condensers, to provide *two* steam ejectors, each of capacity as obtained by use of these curves, and, for jet condensers, two ejectors of a total capacity as indicated by curves, as the jet curves are based on *maximum* air entrainment in injection water and an appreciable amount of air is carried out through the removal pump.

For dry-air pumps of the reciprocating-piston type, the ratio of piston displacement to volume of condensate is approximately as follows:

20 to 1 for 26-in. vacuum	40 to 1 for 28-in. vacuum
30 to 1 for 27-in. vacuum	50 to 1 for 29-in. vacuum

The curves in Fig. 482 give a comparison between the performance of an hydraulic air pump with 70 deg. fahr. hurling water temperature and

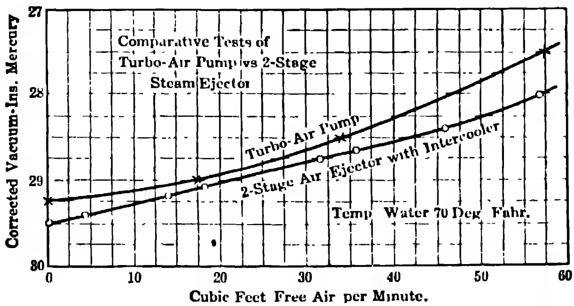


FIG. 482. Comparative Tests of Turbo-air Pump vs. Two-stage Steam Ejector.

that of a steam-jet ejector with inter-cooler. These curves are applicable only to the particular designs tested but serve to show the general characteristics of the two types. The curves in Fig. 483 are of interest in showing the relation between steam consumption and air-removal capacity of a "Radojet" air pump with and without intercooler and with both stages in operation when furnished with dry steam at 100 lb. gage at each stage.

The absolute initial pressure at the jet should be approximately eight times the maximum allowable back pressure. Further increase in initial pressure fails to increase the vacuum and merely increases the steam consumption.

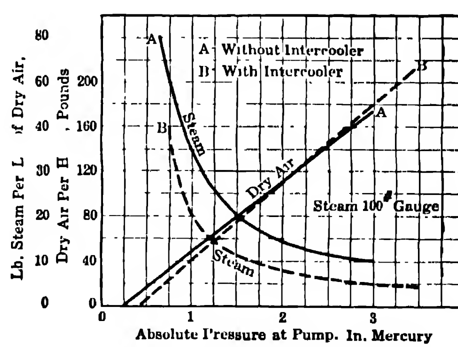


FIG. 483. Performance of Radojet Air Pump when Handling Atmospheric Air.

The Size of Dry-vacuum Pump to Employ in a Given Case. Trans. A.S.M.E., Vol. 44, 1922, p. 437; Power, July 8, 1924, p. 52.

Dry-vacuum Pump Capacity Tests: Power, June 14, 1921, p. 990.

287. Circulating Pumps. — In general, circulating pumps for surface condensers are designed for large capacity against comparatively low heads. Except in some of the older stations, these pumps are of vertical

or horizontal centrifugal type. For heads of 25 ft. or more, single impeller pumps are recommended, driven either by turbine or motor or both as may be dictated by the heat balance of the plant. In large installations multi-rotor pumps are usually installed so as to accommodate high-speed operation to low heads of 15 to 25 ft. Steam-turbine driven pumps give the highest flexibility for condenser operation on account of the ease with which the speed can be changed to take care of fluctuations in head; or, under constant head conditions, to increase or decrease the quantity of water required. This is also true of the variable-speed motor drive, but until quite recently centrifugal circulating pumps were operated with constant-speed motors with no attempt to regulate the supply of water. Some of the more recent installations employ two constant-speed pumps on a single condenser having a divided water-box construction. These pumps are provided with discharge valves as well as a by-pass valve so that either pump may be used to supply water to the entire condenser, or each pump may supply water to one-half of the condenser independently of the other half. The economies effected by this combination have not come up to expectations. In the very latest large central stations, there are two pumps to a single condenser, each equipped with a variable-speed motor. This arrangement gives approximately the same efficiency at reduced flow as at maximum flow and effects a considerable saving in power over the single-pump installation aside from increased flexibility of operation.

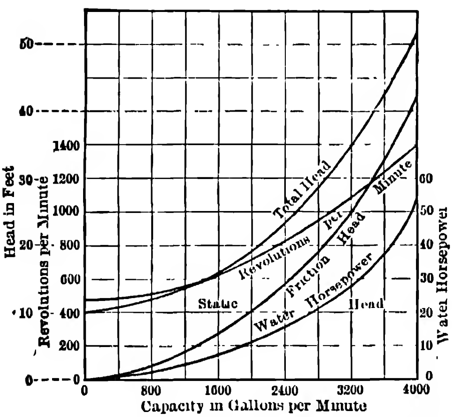


FIG. 484. Performance Curves of a Typical Circulating Pump Installation.

The power required by the circulating pumps is the largest item of the condenser auxiliaries and, therefore, every effort should be made to reduce the pumping head and the required quantity of circulating water to a minimum. Where it is possible to seal the circulating-water discharge pipe, the system operates as a siphon and the static head is the difference in level of intake and discharge tunnels. Where the discharge head cannot be sealed, the static head is the difference in level of intake water and the top pass in the condenser. The total head pumped against in any case is the sum of the static head (suction plus discharge, friction head lost in the condenser and piping and velocity head). The brake

horsepower necessary to deliver water by any type of pump is

$$\text{Br.hp.} = WH/33{,}000E \qquad (262)$$

in which

W = weight of water delivered, lb. per min.,
H = total head, ft.,
E = mechanical efficiency of the pump.

TABLE 87

LARGE CIRCULATING PUMP INSTALLATIONS

(1921-24)

Station	Rated Capacity Turbine Kw	Number of Pumps per Condenser	Maximum Capacity of Each Pump G P M	Rated Hp of Pump Drive	Type of Drive
Barbados .	20,000	1	24,000	175	Constant-speed motor
Cahokia ...	30,000	2	32,000	250	Variable-speed motor
Calumet .	30,000	1	55 000	800	Constant-speed motor
Delaware....	30,000	2	37,500	430	1 Motor, 1 Duplex
Hell Gate..	40,000	1	35,000	350	Constant-speed motor
Hudson Ave...	50,000	2	50,000	700	Constant-speed motor
Northeast..	30,000	2	30,000	1 200	2-speed motor
do.... .				1-200	Constant-speed motor
Lakeside....	20,000	2	18,000	1-170	Constant-speed motor
				1-170	Turbine
Marysville	20,000	1	40,000	253	Variable-speed motor
Steel Point	10,000	1	16,250		Geared turbine
Wabash River.	20,000	2	25,000	150	Constant-speed motor
Waukegan	25,000	1	35,000	300	2-speed motor
Weymouth	30,000	2	39,000	300	Constant-speed motor

The static head of course remains constant, other conditions being the same, for all rates of flow, but the friction increases approximately with the square of the quantity pumped. This is illustrated in Fig. 487. The friction through the condenser tubes may be calculated by means of equation (279) and the friction through the pipe and fittings as shown in paragraph (309).

Example 78. — Calculate the power required to drive the circulating pump for a surface condenser installation when operating under the following conditions: Maximum capacity of main turbine 10,000 kw., water rate of turbine and auxiliaries 15 lb. per kw-hr., ratio of cooling water to condensate 60, suction head 5 ft., friction head 20 ft., static discharge head 15 ft., pump efficiency 78 per cent, pump discharge velocity 15 ft. per sec.

Solution. — The velocity head = $V^2/2g = 15^2/64.4 = 3.5$ ft.
From equation (262),

$$\text{Br.hp.} = \frac{15 \times 10{,}000 \times 60\,(5 + 20 + 15 + 3.5)}{60 \times 33{,}000 \times 0.78} = 254 \text{ (approx.)}.$$

If the pump is motor-driven, allowing an overall motor efficiency of 85 per cent, the pump will require

$$254 \div (10{,}000 \times 1.34 \times 0.85) = \text{0.022 or 2.2 per cent of the main generator output.}$$

Control of Circulating Water: Report of Prime Movers Committee, N.E.L.A., 1923, Part B, p. 89.

288. Centrifugal Boiler-feed Pumps. — In power plants having capacities over 500 b.hp., direct-acting and power-driven triplex boiler-feed pumps have been largely superseded by multi-stage centrifugal pumps. For plants under 500 hp. the direct-acting pump offers the advantage of low first cost and ease of operation. In the modern plant, centrifugal boiler-feed pumps are driven by motors, steam turbines, or both, depending upon the method of establishing the heat balance. Turbine-driven boiler-feed pumps are usually equipped with a constant-pressure, and if desired, excess-pressure or follow-up governor. The admission valve can be set to keep the pressure on the delivery side constant, regardless of capacity pumped, or at some predetermined pressure in excess of that of the steam. The regular turbine governor is adjusted so that it does not function until a speed greater than can be obtained with the follow-up is reached. With some constant-speed motor drives, the feed pumps are fitted with a large pressure-reducing valve so connected that it acts as an excess-pressure throttling valve on the discharge of the pump, while in others an unloading valve is used which allows sufficient water to by-pass from the discharge to the suction of the pump, thereby maintaining a constant discharge pressure. A very satisfactory combination from the operating standpoint is to use both turbine- and motor-driven pumps, the motor-driven pumps to operate at full load and the turbine-driven pumps to operate in parallel for pressure regulation. Slip-ring induction and direct-current motor controls are also available which automatically regulate the speed of the motor-driven boiler-feed pumps to maintain a pressure in the boiler feedwater main at a predetermined amount above that in the boiler. Centrifugal boiler-feed pumps require from a fraction to 5 per cent of the boiler steam generated, depending upon the load, efficiency of the boiler unit, nature of the drive and disposition of the exhaust steam if turbine-driven.

The characteristics for a boiler-feed pump are similar to those shown in Fig. 462. The drooping head delivery characteristic makes it impossible for the pump to overload the driving motor.

Example 79. — Calculate the power required to drive the centrifugal feed pump for a turbine installation when operating under the following conditions: Maximum output of main turbine 10,000 kw., water rate (including auxiliary steam) 16 lb. per kw-hr.; boiler pressure 200 lb. gage.

Solution. — When specific figures are not available it is customary to assume 25 per cent of the boiler pressure as the friction head, whence $H = (200 + 50)\ 2.6 = 650$ ft. (2.6 = ft. of water at boiler temperature corresponding to 1 lb. per sq. in.). Assume a pump efficiency of 65 per cent.

From equation (262),

$$\text{Br.hp.} = \frac{16 \times 10{,}000 \times 650}{60 \times 33{,}000 \times 0.65} = 81.$$

If the pump is driven by a turbine and the latter uses 40 lb. of steam per b.hp-hr., the pump will require

$$\frac{81 \times 40}{160{,}000} = 0.02 \text{ or 2 per cent of the total weight of steam generated.}$$

If the pump turbine exhaust is used for feedwater heating, the pump will require only 0.3 per cent of the total steam generated.

If the pump is motor-driven, allowing an overall motor and line efficiency of 85 per cent, the pump will require

$81 \div (10{,}000 \times 1.34 \times 0.85) = .0071$ or 0.71 per cent of the main generator output at rated load.

289. Condensate or Hotwell Pumps. — The centrifugal pump is now quite universally used for pumping the condensate from surface condensers. Condensate pumps must deliver water against the head corresponding to the vacuum, plus the friction head and the static head. The pump cannot create a vacuum sufficiently greater than the vacuum in the condenser to draw water into the impeller by suction, therefore the condensate should be supplied under a head of three or four feet or more. If the head on the suction side is less than this, the pump " cavitates " or becomes vapor bound and is unable to remove the water. Condensate pumps are built in single-stage and two-stage types. These pumps are ordinarily operated without automatic control and are permitted to operate at constant speed. In the modern central stations these pumps are in duplicate. The power required to operate the pump may be calculated with the aid of equation (263).

Example 80. — Calculate the power required to drive the condensate pump for a turbine installation when operating under the following conditions: Maximum output of main turbine 10,000 kw., water rate (including steam required to operate auxiliaries) 15 lb. per kw-hr., vacuum 28 in. referred to a 30-in. barometer.

Solution. — Suction head corresponding to 28 in. of mercury = 31 ft. Assume a friction and discharge head of 29 ft.; efficiency 50 per cent. Substituting these values in equation (262),

$$\text{Br.hp.} = \frac{10{,}000 \times 15 \times (31 + 29)}{60 \times 33{,}000 \times 0.5} = 9.4 \text{ (approx.)}.$$

290. Air Lift. — The air lift is a simple arrangement of piping whereby water may be raised by means of compressed air. There are no working parts, and no valves are employed except to regulate the supply of air. Its particular field of application lies in pumping water from a number of scattered wells, and on account of the total absence of working parts it is peculiarly adapted to handling water containing sand, grit and the like. The device consists of a partially submerged water pipe and air supply, variously arranged as in Fig. 485 (*A*) to (*D*). Compressed air, forced

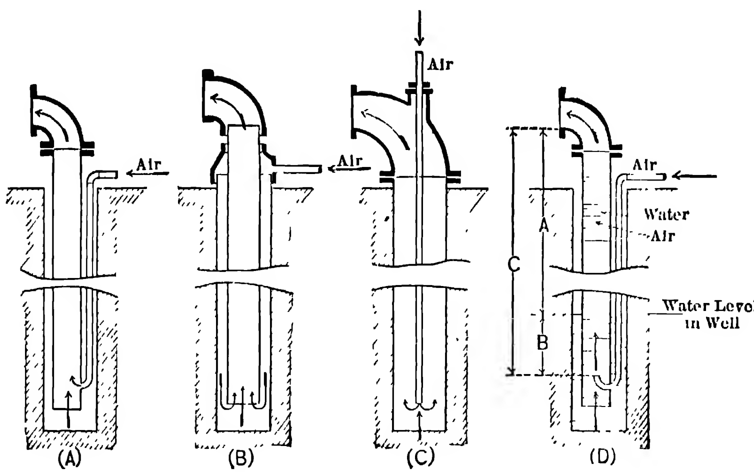


FIG. 485. Various Arrangements of the "Air Lift."

into the water pipe at or near the bottom, decreases the density of the column, and the difference in weight between the solid column of water *B* and the air-water column *A* causes the flow. The successful operation of this device depends upon the ratio of the depth of submersion *B* to the total head *C*.

The quantity of air necessary to operate an air lift may be closely approximated from the equation (see Prac. Engr. U. S., April 1, 1912, p. 354)

$$V = L \div \log \frac{S + 34}{34} \times C \tag{263}$$

in which

V = cu. ft. of free air per gal.,
S = actual submergence in ft.,
C = coefficient determined from experiment.

The actual submergence S may be determined from the relationship

$$S = LS_p/l_p \tag{264}$$

in which

L = actual lift in ft. (A, Fig. 485),
S_p = submergence percentage (100 B/C, Fig. 485),
l_p = lift percentage (100 A/C, Fig. 485).

The coefficient C may be approximated as follows:

$$C \doteq 255 - 0.1\ L. \tag{265}$$

For the air pressure required for any lift and any percentage of submergence, it is convenient to divide the actual submergence in feet by 2 to get the gage pressure in lb. This gives enough pressure in excess of that due to water head to allow for the pipe friction and other losses.

The efficiency (" water " hp. divided by " air " hp.) varies from 30 to 50 per cent, increasing as the ratio B/C increases from 0.55 to 0.85. (*Engineer, U. S.*, Aug. 15, 1904, p. 564.) A number of tests give efficiencies (" water " hp. divided by i.hp. of steam cylinder) varying from 20 to 40 per cent. The hp. required to compress one cu. ft. of free air to different pressures per sq. in., as determined from actual practice, is approximately as given in Table 87*a*.

TABLE 87*a*

Pressure in Pounds	Hp. Required to Compress 1 Cubic Foot	Pressure in Pounds	Hp. Required to Compress 1 Cubic Foot
176	0.434	60	0.159
140	0.376	45	0.145
100	0.261	30	0.121
80	0.189		

Air Lift: Power, Nov. 23, 1920, p. 818; Apr. 17, 1923, p. 591; Jan. 30, 1923, p. 177; May 6, 1919, p. 692; Bul. No. 1265, 1924, Univ. of Wis.

291. Pump Governors. — Steam-driven pumps are readily adapted to automatic control since it is only necessary to regulate the speed by throttling the steam supply. Figure 486 shows a section through a Fisher pump governor illustrating the general principles of **constant-**

pressure control on pumps of this class. It embodies a pressure-reducing valve in the steam supply pipe of the pump, actuated by the slight variations in water pressure. When the demand for water increases, the pressure in the discharge pipe tends to decrease, and this drop in pressure (transmitted to the pump governor through opening *D*) causes more steam to be admitted, which increases the speed of the pump. The governor is connected to the steam inlet of the pump at *B* and the steam enters at *A*. The double-seated balanced valve *C* regulates the supply of steam to the cylinder by the amount it is raised from the seat. The valve is held open by spring *G*, the compression of which may be regulated by han wheel *K*. The water pressure from the discharge pipe acts on piston *F*, and tends to overcome the resistance of the spring. The difference in pressure between the water and the spring determines the position of valve *C*. The spring tension is adjusted by means of the hand wheels.

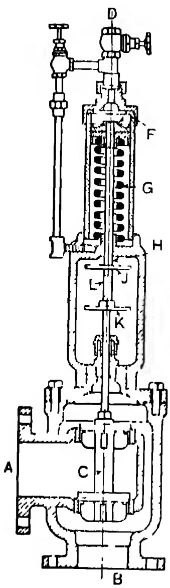


FIG. 486. Constant - Pressure Steam - Pump Governor.

For maintaining a constant pressure in the suction line of a steam-driven vacuum pump, the spring-loaded piston in the governor is replaced by a lever-weighted diaphragm.

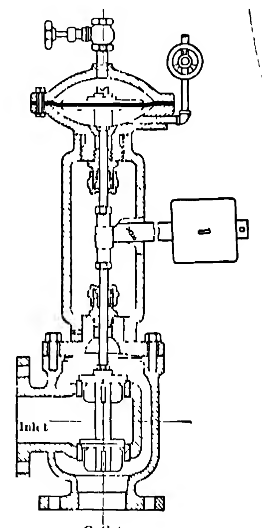


FIG. 487. Excess-pressure or Follow-up Steam-Pump Governor.

Figure 487 shows a section through a Fisher **excess-pressure** or **follow-up** type of steam-pump governor which will maintain a constant difference in pressure irrespective of the variations in steam pressure or the capacity of the pump. Governors of this type are usually installed in connection with boiler-feed pumps. They differ from the constant pressure type only by the substitution of a lever-weighted diaphragm for the spring-actuated piston and in the connection of the steam pressure to the under side of the diaphragm and of the feed line pressure to the upper side. The diaphragm, therefore, has to support only the excess pressure necessary to overcome the weight and lever. This type of governor is readily applied to a steam-turbine-driven centrifugal pump.

Figure 488 shows a section through an excess-pressure governor incor-

porated in the design of a Lee steam turbine. The water end of the governor *W* is piped to the discharge line of the pump and the steam end *S* is connected to the steam line. Pressure is thus introduced to the closed space at each end of the governor body and acts upon the diaphragms *D*, *D*. These are connected by a diaphragm spacer which, through the lever *L*, actuates the governor lever *G* and in turn the governor valve *V*. The predetermined excess-pressure is produced by a coil-spring. The speed of the turbo-pump, therefore, will be held at a point where the water pressure equals the steam pressure plus spring pressure irrespective of the variations in pressure of the water or steam.

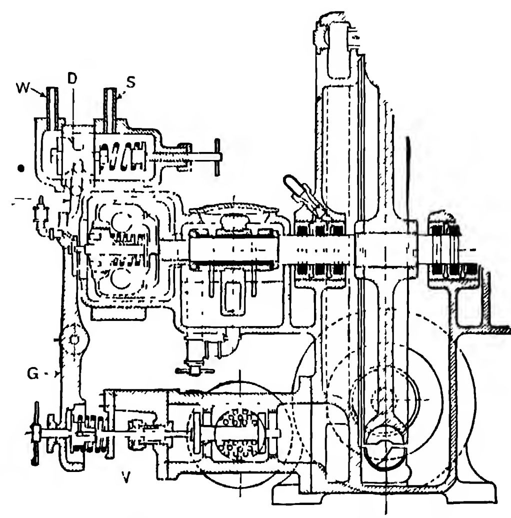


Fig. 488. Excess-pressure Turbo-pump Governor.

292. Feedwater Regulators. — In the great majority of the older steam plants, the supply of feedwater to the boiler is controlled by hand. By opening or closing a regulating valve in the pump discharge line, the supply is throttled to meet the boiler requirements. Since it is practically impossible to manipulate the valve so that the water will flow into the boiler as fast as the steam is driven off, the flow is more or less intermittent, the water level ranging from maximum to minimum. Practically all of the large modern central stations and many of the smaller installations are equipped with automatic regulators, not only to insure continuous feeding of the boiler at the proper rate, but to dispense with the constant attention necessary for hand control. Feedwater regulators depend upon the fluctuations of the water level in the boiler for their primary control.

Figure 489 shows a section through a **Stets boiler-feed controller** illustrating the **float-lever** type. The float chamber is connected to the steam space of the boiler or water column and to the lower water-gage in such a manner that the mean water level in the chamber is in a line with that in the boiler. A copper float, rising and falling with the water level, actuates, through the agency of suitable levers, a balance feed valve and reduces or increases the flow of water to the boiler. A

fixed relation is maintained between water level and feed valve opening. As the working parts are all in the pressure space, very little stuffing box friction is interposed between the float and feed valve. As the float contains a small amount of alcohol, internal pressure when in operation is practically the same as the external pressure of the steam, and therefore the ball is subjected to very little stress. The linkage and valve openings are designed to give a continuous flow of water in gradually increasing rate from high to the low water level limit.

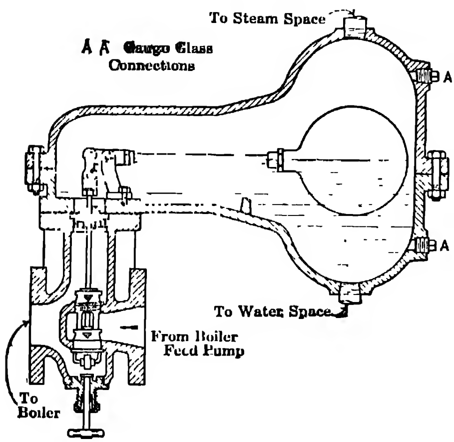


FIG. 489. Stets Boiler-Feed Controller.

Figure 490 shows a section through the actuating end or generator of the **"S-C" regulator** illustrating the **thermo-pressure** type. The generator consists of an inclined seamless brass tube *S* connected through suitable fittings to the steam and water space in the boiler. The level of the water in this tube will correspond to that in the boiler. Tube *S* is surrounded by vessel *T* which is closed at both ends and does not communicate with the boiler. Vessel *T* is equipped with thin bronze fins to carry away heat. This vessel is filled with water, which always remains in the system, and is connected through flexible tubing *F* to the top of a diaphragm-controlled balanced valve in the feed pipe. When the water in the boiler is at its highest permissible level, the tube *S* is filled with boiler water, the temperature of the independent water body in vessel *T* is comparatively low, and the feed valve is closed. As the level of the water in the boiler is lowered by the discharge of steam through the boiler nozzle, the level in tube *S* drops correspondingly and the upper end is filled with steam. This steam gives up heat to the water in vessel *T* and causes it to expand. The pressure created by this expansion is trans-

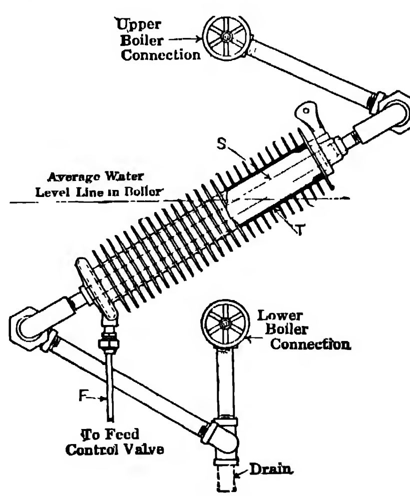


FIG. 490. "S-C" Feedwater Regulator.

mitted to the diaphragm chamber of the feed valve and opens it a proportionate amount. The lower the water level in tube S, the greater the surface exposed to steam and hence the higher the pressure developed in the generator. Conversely, as the water level rises, less surface is exposed to steam, and through the action of the radiating fins, the temperature, and consequently the pressure in chamber T, is reduced. The lower connection of tube S is trapped so that the water in the tube will be at a much lower temperature than that in the boiler.

Figure 491 shows the general arrangement of a **Copes feedwater regulator** illustrating the **thermo-expansion** type. The regulator is actuated by the expansion and contraction of a heavy metallic tube mounted on a base and connected by a lever and strut to a control valve in the feed line. The expansion tube is set in an inclined position opposite the water level in the boiler, and is connected to the steam and water spaces as shown. The water level moves in the expansion tube as it does in the boiler. The upper end of the expansion tube, being filled with steam, is at steam temperature. The lower end, however, is slightly above room temperature because the water which it contains gradually cools off by radiation. At normal load, with the water level at position " 2," Fig. 491, half of the expansion tube is filled with steam and half with water, and the tube assumes the length that opens the central valve to position " 2." As the load gradually increases, the water level falls and the tube is filled with steam. This expands the tube and causes the valve to open proportionately, as shown by position "1."

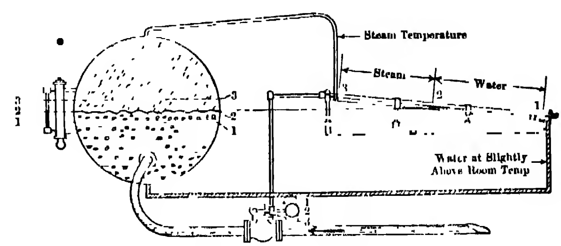


FIG. 491. Copes Feedwater Regulator.

For satisfactory operation, all feedwater regulators should be used in connection with suitable pump governors.

PROBLEMS

1. A direct-acting duplex boiler-feed pump uses 125 lb. steam per i.hp-hr. Initial steam pressure 115 lb. abs., feedwater temperature 180 deg. fahr. What per cent of the total steam generated by the boiler is necessary to operate the pump?

2. A triple-expansion pumping engine delivers 30,310,000 gallons of water in 24 hours against a head of 61 lb. per sq. in., initial steam pressure 200 lb. abs., developed hp. 800, water rate 10.33 lb. per br.hp-hr., steam initially dry. Required the duty per 1000 lb. of dry steam and per million B.t.u.

3. Determine the cylinder dimensions of a direct-acting single-cylinder feed pump suitable for a 500-hp. boiler, maximum overload 100 per cent, boiler pressure 115 lb. abs., feedwater temperature 70 deg. fahr.

4. Required the probable i.hp. when operating at maximum capacity.

5. Which is the more economical in heat consumption as a pump, an injector or a motor-driven triplex power pump? Boiler pressure 100 lb abs., feedwater supply 60 deg. fahr., injector delivers 16 lb. of water per lb. of steam, overall efficiency of pump and motor 60 per cent.

6. Approximate the cylinder dimensions of a wet-air pump for a 750-hp. engine using 16 lb. steam per i.hp-hr., initial pressure 150 lb abs., vacuum 26 in. (barometer 30 in.), dry steam at admission, initial temperature of injection water 70 deg fahr.

7. Required the horsepower necessary to operate a centrifugal circulating pump for a surface-condenser installation using 1000 gallons of water per minute, total head pumped against 50 ft, initial temperature of circulating water 70 deg fahr.

8. If a motor-driven centrifugal pump (head and temperature conditions as in Problem 7) is installed in connection with a 1000-hp. engine and the ratio of cooling water to condensed steam is 30 to 1, required the per cent of main engine power necessary to operate the pump.

9. If the pump in Problem 8 is driven by a steam engine using 50 lb. steam per hp-hr. and the exhaust is used for heating the feedwater, required the per cent of main engine heat supply necessary to operate the pump. Main engine initial pressure 150 lb. abs., vacuum 26 in (barometer 30 in.), circulating-pump engine initial pressure 100 lb. abs., back pressure 16 lb. abs. Assume dry steam at admission in both cases.

CHAPTER XV

SEPARATORS, TRAPS, AND DRAINS

293. General. — While separators, traps, and drains appear to be insignificant factors in the steam power plant, the economy and physical operation of the station are largely dependent upon their correct installation and proper functioning.* High-pressure saturated steam flowing from boiler to prime mover or auxiliaries always contains more or less water, due either to priming or foaming in the boiler or condensation in the pipe lines. Aside from increased conductivity due to the higher density which increases the condensation losses, the water content may cause water hammer in the pipe line or even destruction of the prime mover and auxiliaries if they are of the reciprocating type. With steam turbines, the moisture content reduces efficiency and causes excessive erosion of the blades. Practically all of the scale-forming elements carried along with steam are in the moisture content, so that elimination of the moisture will remove these impurities and prevent them from fouling the superheater coils, clogging the valves and fittings, and cutting the turbine blades. An efficient separator, such as the "Tracy Steam Purifier," placed within the boiler and taking the place of the customary dry pipe, will insure perfectly dry steam delivery to the superheater or to the main steam header. **Drip pockets,** which are in reality steam separators, placed at suitable points in the pipe line, will remove a considerable portion of the water entrainment, while a correctly designed **steam separator,** located at the engine throttle, will eliminate all but a trace of the moisture carried to that point. Exhaust steam from reciprocating engines and pumps contains not only moisture due to cylinder condensation and that resulting from work done by the steam, but a varying amount of the oil used for lubrication. The greater part of the oil appears as an emulsion with the moisture in the exhaust, so that elimination of the moisture will automatically remove the oil. Where oil-free exhaust is necessary for industrial processes or where the exhaust is to pass into a low-pressure turbine and moisture-free steam must be provided to prevent excessive blade corrosion, an **oil eliminator** or **separator** will remove nearly all the emulsified condensate. The moisture trapped from the high-pressure lines, steam jackets, receiver coils or other high-pressure appliances has a

considerable heat content and, unless the plant is so small that this heat is of little consequence, it is customary to reclaim the condensate and return it directly to the boiler, heater, or hot-water storage tank. This duty is performed by **steam traps,** or similar automatic return apparatus.

Theoretically, there is no need for live steam separators, traps, or drains in superheated steam lines where the temperature is never less than saturation, but in starting up there is always some condensation, and at times there is a possibility of the superheater becoming flooded and of slugs of water being carried over into the piping system. To provide against the water reaching the prime mover, drip pockets are placed in the line, suitable drains at the superheater, and frequently a steam separator at the throttle. Whenever it is desired to use small piping with steam at high velocities and at the same time reduce the velocity at the throttle or provide a damping effect for pulsation, a **receiver-separator** placed close to the throttle will effect the desired result.

Separators, traps, and drains are designed for all grades of service, high pressures, medium or low pressures and vacuum. Some of the more important appliances will be briefly described.

294. Live Steam Separators. — Any pocket placed in a horizontal run of piping will remove all or a part of the moisture entrained with the steam, provided the velocity is not such as to carry all the water in suspension. For velocities below 10 ft. per sec., practically all the water will collect in a pocket having a diameter of opening equal to that of the pipe, but since this is far below the minimum velocity allowed in most stations, a plain drip pocket will remove only a portion of the water entrainment. In order to remove the greater part of the moisture, the separating vessel must be designed so that one or more of the following principles are employed.

1. **Reverse-current.** The direction of the flow is abruptly changed, usually through 180 deg. This causes the water in the steam, on account of its greater specific gravity, to be thrown into a receiving vessel, while the steam passes on in a reverse direction.

2. **Centrifugal force.** A rotary motion is imparted to the steam, whereby entrained water particles are eliminated by centrifugal force.

3. **Baffle-plates.** The flow is interrupted by corrugated or fluted plates, to the surfaces of which the water particles adhere and from which they fall by gravity to the well below.

4. **Mesh.** The separation is brought about by mechanical filtration through screens or meshes.

The following outline shows the classification of typical separators, in accordance with the above principles:

Live-steam separators.	Reverse-current . . .	Hoppes. Stratton.
	Centrifugal	Keystone. Mosher. Robertson. Bundy.
	Baffle-plate.	Austin. Tracy.
	Mesh.	Direct. Potter.

A series of tests made at Armour Institute of Technology in 1905 on a number of separators showed that the *efficiency of separation decreased as the velocity of the steam increased.* At the low velocity of 500 ft. per min. all separators were equally efficient (about 99.8 per cent), at a velocity of 5000 ft. per min. several had little effect, and at a velocity of 8000 ft. per min. only one gave efficient results. For this reason, it is better to err in installing too large a separator than one which is too small. Furthermore, the pressure drop through the separator increases approximately as the square of the velocity and may become excessive at velocities over 100 ft. per sec.

Reverse-current Steam Separators. — Figure 492 shows a section through a Hoppes steam separator and illustrates the principle of reverse-current separation. Steam may flow through in either direction. Both the inlet and outlet ports are surrounded by gutters C, C, partly filled with water, which intercept the moisture following the surface of the pipe, while the downward plunge of the steam throws the entrained water to the bottom of the separator. The condensation is carried from the troughs by pipe P to the well below, from which it is trapped at D in the usual way. The velocity of the steam in passing through this separator is greatly reduced to prevent the steam from taking up the water in the bottom of the well. This is brought about by increasing the area of the passage through the separator.

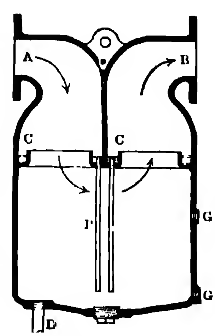


FIG. 492. Typical Reverse - current Separator. (Hoppes.)

Figure 493 gives a sectional view of a Stratton separator, which, though primarily of the reverse-current type, embodies also the principle of centrifugal force. The separator consists of a vertical cast-iron or cast-steel cylinder with an internal central pipe C extending from the top downward for about half the height of the apparatus, leaving an annular space between the two. The current of steam on entering is deflected by a curved partition and thrown tangentially to the annular space at the side,

near the top of the apparatus. It is thus whirled around with all the velocity of influx, producing the centrifugal action which throws the particles of water against the outer cylinder. These adhere to the surface, so that the water runs down continuously in a thin sheet around the

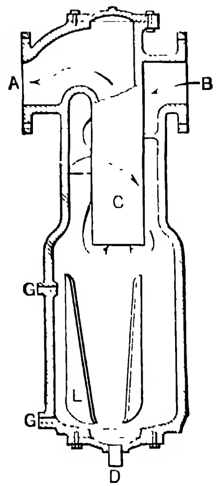


FIG. 493. Stratton Steam Separator.

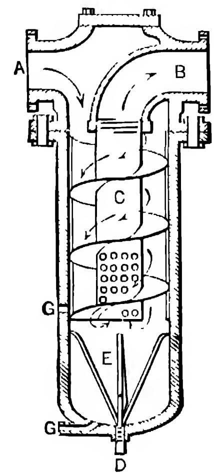


FIG. 494. Combined Reverse-current and Centrifugal Separator. (Keystone.)

outer shell into the receptacle below. The steam, following in a spiral course to the bottom of the internal pipe, abruptly enters it, and passes upward and out of the separator without having once crossed the stream of separated water. The rapid rotation of the current of steam imparts a whirling motion to the separated water which tends to interfere with its proper discharge from the apparatus. The separator has therefore been provided with wings or ribs *E* projecting at an acute angle to the course of the current, which have the effect of breaking up this whirling motion and allowing the water to settle quietly at the bottom, whence it passes off through the drain pipe *D*.

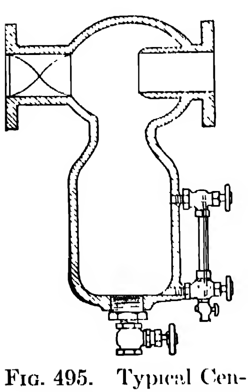


FIG. 495. Typical Centrifugal Separator. (Swartout.)

Centrifugal Steam Separators. — Figure 495 shows a section through a Swartout centrifugal separator. The helix in the inlet opening imparts a whirling motion to the steam without reducing its velocity. The water particles are thrown against the wall by centrifugal force, while the steam passes on in a dry state.

Baffle-plate Steam Separators. — Figure 496 gives an interior view of a

Bundy separator and illustrates the application of baffle plates for live-steam separation. This separator consists of a rectangular cast-iron casing with a cylindrical receiver beneath it. Directly across the steam passage are baffle-plates corrugated for the reception of entrained water. The plates consist of vertical castings, each containing a main artery or channel which leads directly to the receiver. The fronts of the plates are flat, with a series of recesses sloping inwards and downwards, terminating in an opening of capillary size leading to the main artery. The plates are staggered, so that the steam must impinge against all of them in its passage. The particles of water adhere to the plates, collect, and fall by gravity into the receiver. The flanges at the bottom constrict the opening of the reservoir so as to prevent the steam from picking up any portion of the water.

Figure 497 shows a section through an Austin separator and illustrates another class embodying the fluted baffle-plate principle. The steam in

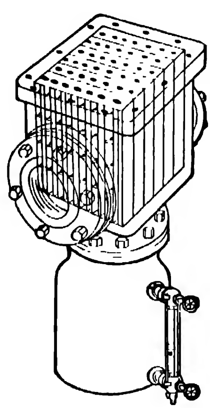


FIG. 496. Bundy Steam Separator.

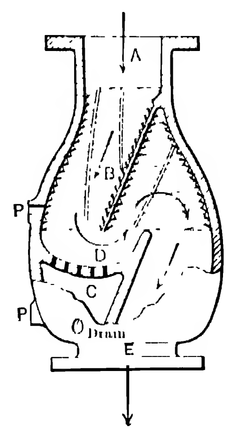


FIG. 497. Austin Steam Separator. (Fluted Baffle.)

passing through the chamber impinges against the fluted baffle-plate B. The moisture adheres to the surfaces, collects and trickles along the corrugations to the bottom of the well. These corrugations are formed in such a manner that the steam cannot come in contact with the water particles after they have been once eliminated. A perforated diaphragm D prevents the water in the well from coming in contact with the steam. The current of steam is also reversed, thus giving additional separating properties to the apparatus.

The **Tracy steam purifier** or " dry pipe " consists essentially of a number of rows of baffle plates placed inside the boiler (taking the place of the

customary dry pipe) in such a manner that the steam in flowing through the narrow channels impinges against the surface and leaves the moisture adhering to them. The velocity through the channels is very low (750 ft. per min.) so that the moisture is not picked up again by the steam. The moisture gravitates to the bottom of the chamber and is discharged to waste.

Mesh Separators. — Figure 498 shows a section through a "direct" separator, illustrating the principle of mesh separation. These separators are made with steel bodies and cast-iron heads and bases, in all sizes up to 6 in. inclusive, the larger sizes being constructed of cast-iron or boiler plate. The cone C, perforated lining E, and diaphragm S are made of cold-rolled copper; the cone O is a substantial gray-iron casting, resting on three cast-iron supports hooked over the top of inner pipe as indicated. The method of operation is as follows: The accumulated moisture around the walls of the steam pipe is caught by the upper edge of cone C and carried down back of lining E to the water chamber. The current of steam entering the separator impinges upon the conical surface, which is composed of solid plate O covered with sieve S, through which water may freely pass but from which it cannot readily escape. Passing through the sieve and depositing on the solid surface of the cone O, this water is carried by conductors P to the water chamber. Perforated lining E permits the moisture content of the steam to pass through the opening to the water below and prevents it from coming in contact again with the current of steam. A trough is provided at the lower edge of the inverted cup, which leads all the water that may adhere to it to the water chamber. The steam flows through the passages indicated by arrows and is subjected to a whip-snapping action which tends to throw off any remaining moisture. The perforated plate D prevents the steam from picking water out of the water chamber.

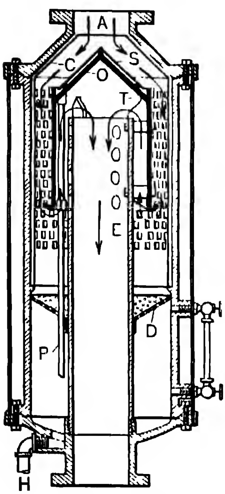


FIG. 498. "Direct" Steam Separator.

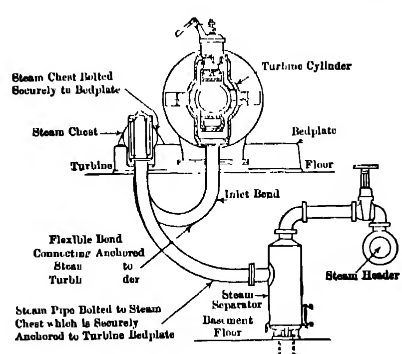


FIG. 499. Separator Applied to a Steam Turbine — Basement Header.

Figures 499 and 500 show typical installations of live-steam receiver-separators to steam turbines. With reciprocating steam engines, it is

customary to place the separator close to the main throttle without intermediate piping.

295. Exhaust-steam Separators and Oil Eliminators. — The function of an exhaust-steam separator is the removal of cylinder oil from the steam exhausted by engines and pumps. In plants where exhaust steam is used for heating, it is quite essential to remove the oil from the steam before it enters the heating system, for the oil not only reduces the efficiency of the radiators by coating them with an excellent non-conducting film but is an element of danger to the boiler itself. In surface-condensing plants the separator will prevent the oil from fouling the condenser tubes and those of the vacuum heater (if one is installed); this is an important factor, since the oil or grease lowers the efficiency of the heat transmission.

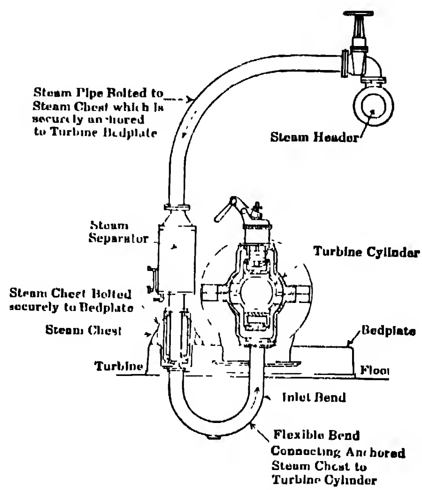


FIG. 500. Separator Applied to Steam Turbine — Overhead Steam Main.

In a general sense, a live-steam separator is also an oil eliminator, and all the separators previously described perform this function to a certain extent, since the underlying principles governing the elimination of oil from exhaust steam are similar to those employed in removing water from steam. Most of the separators described above are also designed in lighter form, as oil eliminators, but by far the greater number are based on the fluted baffle-plate principle, of which the Austin, Bundy, Cochrane, Utility, Crane, and Keiley are well-known examples. This type of oil separator will eliminate a considerable portion of the oil in the steam, provided the baffle-plates or corrugated surfaces are frequently cleaned.

Since the velocity through exhaust steam pipes, particularly in condensing plants, is much higher than with live steam, the separator chamber and the separating surface must be of sufficient size to reduce the velocity; otherwise, separation will be inefficient.

A very successful method of removing oil from steam is to project the steam on to the surface of a body of water. The water may be hot or cold and will hold the oil if it once reaches the surface. It is essential, however, to reduce the velocity of the steam as it passes on its way to the outlet.

The most efficient method of removing all traces of oil is by combined filtration and absorption. A large chamber filled with coke, brick, broken tile, or other absorbing material is placed in series with the exhaust pipe. The steam passing through this chamber is entirely freed from oil and moisture, provided the absorbing material is sufficient in quantity and is replenished as soon as it becomes saturated with oil. The annoyance attending the removal and replenishing of the absorbing material at frequent intervals and the great size of the apparatus are serious drawbacks. An example of this system of purification, in which many of the objectionable features are reduced to a minimum, is the Loew grease and oil extractor, Fig. 501. The exhaust steam enters the chamber at the top, strikes a large deflecting plate shaped like an inverted V, and permits part of the condensation and oil to be drawn off by the drain pipe. The steam then rises and is deflected, as indicated, against a series of shelves filled with fibrous material covered with coarse wire screens. The grease is removed from each shelf by suitable drains. This apparatus is sectional and any number of sections may be added without affecting the rest.

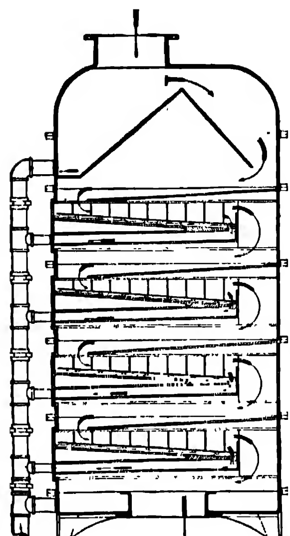


Fig. 501. Loew Grease Extractor.

In a non-condensing plant where the exhaust steam is used for heating purposes, the oil separator is ordinarily placed in the main exhaust pipe just before it enters the heating system. Where several branches enter one main, it is not customary to place a separator in each branch, one large separator located as above being sufficient. Oil separators are also incorporated in the body of exhaust-steam feedwater heaters. If the exhaust is discharged directly to waste, there is no need for a separator except to prevent the oil and water from fouling the roof and polluting the surrounding neighborhood. A separator placed on the roof and at the end of an exhaust pipe is usually designated as an **exhaust head.**

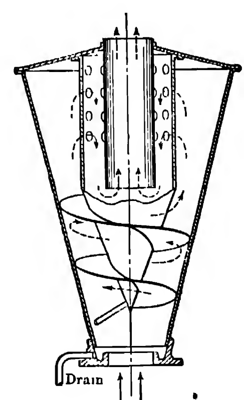


Fig. 502 Typical Exhaust Head.

On condensing plants, exhaust steam separators and oil eliminators are

used only in connection with surface condensers or heaters or when the exhaust is passed into a low-pressure turbine. Where a jet condenser is used, the hotwell itself acts as an efficient oil separator.

296. Disposal of Drips. — The water condensed or otherwise deposited in high-pressure lines should be removed as rapidly as possible to prevent water hammer, blade erosion in turbines, and possible wreckage of piston prime movers and auxiliaries. These **high-pressure drips,** whether collected in small individual drip pockets, separators, or a common reservoir, contain considerable heat, are free from impurities and therefore should be returned to the boiler either directly or indirectly. The same is true for all water of condensation however formed and of whatever temperature, provided the quantity involved is not too small to cover the investment cost of the return equipment, and the quality is not such as to require expensive treatment. In small plants with low load factors, it is frequently more economical to discharge all condensation to waste; in large stations, it may be advisable to save the high-pressure drips and waste only the contaminated low-pressure condensation; but in the large modern central station, provision is made for utilizing all condensate both as regards the heat content and the water itself.

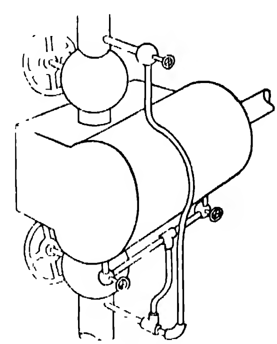


Fig. 503. Simple Method of Draining Drips.

Figure 503 shows one of the simplest means of disposing of the drips in a small piston engine where the quantities involved are too small to warrant conservation. The water collected above the throttle valve and the condensation in the cylinder in starting up are blown directly into the exhaust pipe. This system makes no provision for water carried over in large quantities while the engine is in operation. Possible wreckage of the engine from this cause may be prevented by placing large spring-load snifting valves at the ends of the cylinder.

In small plants with low load factors, and where there is a deficiency of exhaust steam for feedwater heating, the high-pressure drips are frequently piped directly to the heater, the valves in the drip pipes being "cracked" to permit a continuous flow of condensate to the heater. When there is sufficient exhaust to heat the feedwater to a temperature corresponding to that of the exhaust, this may entail a serious loss since it is practically impossible to open the valve so as to let only water escape. In order to prevent the steam from escaping and at the same time permit all condensation to be discharged to any desired point having a pressure less than that of the steam, automatic **steam traps** or **condensation-control**

appliances are employed. Steam traps, in fact, are used for this purpose in practically all plants. In some of the older plants, high-pressure drips were automatically and continuously returned to the boiler by the **steam loop** or **Holly loop,** but few of the modern plants are equipped with these devices.

Low-pressure drips, if contaminated with oil, are either discharged to waste directly or through the agency of steam traps. If the condensate

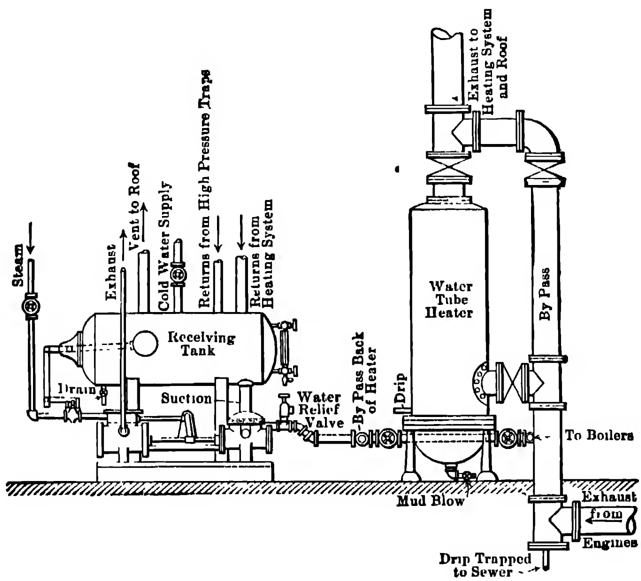


FIG. 504. Returns Tank and Pump.

is pure and fit for re-use as feedwater, it is usually returned to the boiler-feed system by the same means. Where large quantities of water are to be handled, some type of pump is preferred to the trap system.

297. Steam Traps. — Automatic steam traps may be divided into several classes, depending upon the service for which they are intended: (1) **return traps,** in which the water is returned directly to the boiler; and (2) **non-return traps,** in which the condensation is discharged into any receptacle having less than boiler pressure. They may also be classified according to the steam pressures involved as (*a*) high-pressure, (*b*) low-pressure, and (*c*) vacuum. All of these classes may be subdivided into five types according to the principle of operation, viz:

I. Float.
II. Bucket.
III. Bowl.
IV. Expansion.
V. Differential.

CLASSIFICATION OF A FEW WELL-KNOWN STEAM TRAPS

Steam Traps.	Float	Johns-Manville.	
		McDaniel	
	Bucket	Acme	
		Kieley	
	Dump	Bundy	
		Morehead.	
	Expansion	Metal	Columbia.
			Geipel.
		Fluid	Webster.
			Sarco.
	Differential	Flinn.	
		Siphon.	

Float Traps. — Figure 505 shows a section through a Johns-Manville trap, illustrating one of the simplest types of float valves. The float is unattached and free to revolve in any direction, and acts as the valve seat. When completely discharged, the unbalanced pressure in the body of the trap holds the ball against its seat. Water gravitating into the trap lifts the ball off its seat and is discharged through the orifice by the steam pressure acting upon the surface of the liquid. This process continues until the level in the vessel once more brings the ball in contact

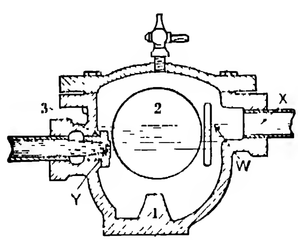


FIG. 505. Johns-Manville Trap.

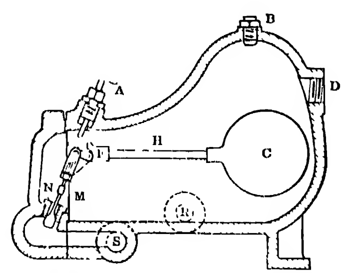


FIG. 506. Typical Lever-float Trap.

with the seat. A gage glass indicates the height of water in the chamber. Figure 506 illustrates a float trap of the lever type. The rising and falling of the float opens and closes the valve in proportion to the rate of discharge.

Unless float traps are well made and proportioned, there is a danger of considerable steam leakage through the discharge valve, due to unequal expansion of valve and seat and the sticking of moving parts. The discharge from a float trap is usually continuous, since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present. When the trap is working lightly, this adjustment is apt to throttle the area and create such a high velocity of discharge as to cause

a rapid wear of valve and seat. This defect is more or less evident in all steam traps discharging continuously. For this reason all wearing parts should be accessible and readily replaceable.

Bucket Traps. — Figure 507 shows a section through an "Improved Acme" steam trap. The water of condensation enters the cast-iron vessel at A, filling the space D between the bucket E and the walls of the trap. This causes the bucket to float and forces valve V against its seat (valve V and its stem being fastened to the bucket as indicated). When the water rises above the edges of the bucket, it overflows and causes the bucket to sink, thereby withdrawing valve V from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water through the annular space H to discharge opening G. When the bucket is emptied, it rises and closes valve V and another cycle begins. By closing valve R the trap is by-passed and the condensation blows directly through passage C to discharge G. The discharge from this type of trap is intermittent.

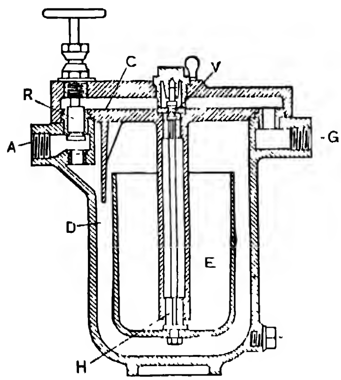


FIG. 507. Typical Bucket Trap.

Dump or Bowl Traps. — Figure 508 shows an elevation of a Bundy bowl trap of the "return" design. The water enters the bowl as indicated, passes through trunnion T and rises until its weight overbalances counterweight W and the bowl sinks to the bottom. As the bowl sinks, arm A, which is a part of the bowl, rises and engages the nuts N on valve stem S and opens valve V, thus admitting live steam pressure on to the surface of the water. The trap then discharges like all others. After the water is discharged, weight W sinks and raises bowl B, which in turn closes valve V, and the cycle begins again. Air valve E is for the purpose of equalizing the pressure in the chamber immediately after discharge. This valve closes just as valve V opens and conversely opens when valve V closes. The air valve is vented to the atmosphere. Bowl traps are necessarily intermittent in their discharge.

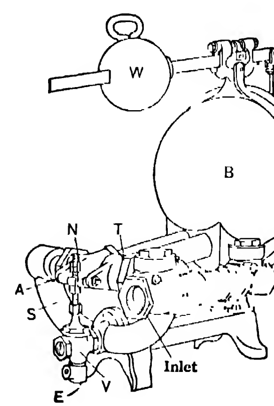


FIG. 508. Typical Dump or Bowl Trap.

Expansion Traps. — Expansion traps may be divided into two groups: (1) those in which the discharge valve is operated by the relative expansion of metals, and (2) those in which the action of a volatile fluid is utilized.

Expansion traps will never freeze, as they are open when cold and all the water drains out before the freezing temperature is reached.

Since traps of this type have little capacity for holding water, 5 to 10 ft. of pipe should be provided between the trap and the pipe to be drained, in order that the condensation may collect and cool.

Figure 509 shows the general appearance of a Columbia expansion trap in which the valve is operated by the expansion of metallic tubes. Water gravitates to the trap through the opening marked " inlet," passes through *brass* pipe *O*, then downward to the main body of the valves and back to outlet valve *C*. Below pipe *O* and parallel to it, is an *iron* rod *S*, at the end of which is the support or fulcrum of lever *R*. The lower end of this lever is connected to the stem of the valve *C*, so that any movement of the lever is communicated to it. When the trap is cold, valve *C* is open and all water of condensation passes out. The moment steam enters the pipe *O*, it expands. The amount of expansion is multiplied several

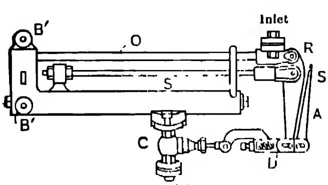


FIG. 509. A Typical Expansion Trap (Lever Type).

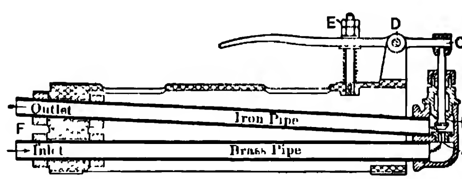


FIG. 510. Geipel Expansion Trap.

times by the action of the lever *R*, so that the movement of the valve is much greater than the expansion of the pipe *O*. The compensating spring *D* prevents the brass tube from damaging itself by excessive expansion. Lever *A* permits the trap to be blown through by hand.

Figure 510 shows a section through a Geipel trap in which the valve is operated directly by the expansion of two metallic tubes and the movement is not multiplied by levers as with the Columbia. The lower or brass pipe constitutes the inlet and is connected to the vessel to be drained; the upper or iron pipe is the outlet for discharge. The two pipes form the sides of an isosceles triangle, the base *F* of which is rigid, while the apex *A* is free to move in a direction at right angles to the linear expansion of the tubes. When cold, the brass pipe is contracted, and the apex, in which the valve seat is placed, is moved down so that the valve is open and the water is discharged. As soon as steam enters the brass pipe, the latter expands and forces the valve seat against the valve. The trap may be

adjusted for any pressure by means of the lock nuts E. When it is desired to blow through, the valve may be operated by hand by pressing the lever.

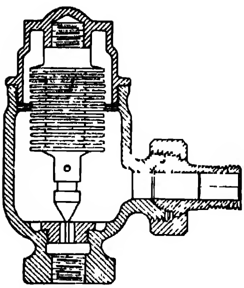


FIG. 511. Webster Silphon Trap.

Figure 511 shows a section through a Webster silphon trap, illustrating the expansion principle in which the motive power is the pressure of a volatile fluid. This style of trap is suited to comparatively low pressures only, and has practically supplanted all others for low-pressure drains. The bellows is formed from a single seamless piece of metal and is filled with a charge of volatile fluid or vapor which boils at a comparatively low temperature. When cold the bellows is contracted and the valve is open. As soon as the temperature for which the silphon is designed is reached, the volatile fluid boils and pressure is developed, extending the bellows and forcing the valve against its seat. When the temperature drops, owing to the cooling of the condensate, the pressure in the silphon is lowered, the bellows contracts and the valve opens.

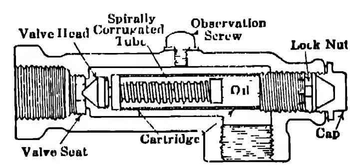


FIG. 512. Sarco Steam Trap.

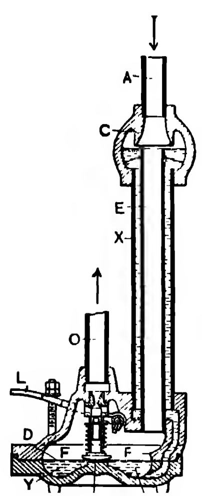


FIG. 513. Flinn Differential Trap.

Figure 512 shows a section through a Sarco steam trap which differs from the Webster in that the actuating fluid (a heavy hydrocarbon oil) surrounds the silphon bellows. This device is suitable for pressures up to 200 lb. per sq. in.

Differential Traps. — Figure 513 shows a cross section through a Flinn differential trap. The column of water X acting on diaphragm D closes valve V. The water entering pipe E and the action of the spring equalize column X and open the valve. Describing the action in further detail, the water of condensation enters at A, fills lower chamber Y, pipe X, and receiving chamber C up to the level of the top of pipe E. This column of water acting on the under side of the diaphragm D forces the valve to its seat against the counter pressure of the spring S. Any additional water that enters the trap overflows through pipe E, filling chamber F and pipe E to a point about midway of its height, where the effect of the column of water in pipe X is balanced. The pressure on

each side of the diaphragm is then equal, the short column in pipe E, aided by the spring, balancing the pressure of the longer column in pipe X. Any further increase in the height of the water in pipe E causes a depression of the valve V, which allows water to escape until the column has fallen to a level a little below the middle of pipe E, when this valve closes again. This action is repeated at intervals according to the quantity of water entering the trap. So long as the water keeps coming in sufficiently large quantities, the valve remains wide open.

Siphon Traps. — Figure 514 gives a general view of a siphon trap, which is much used in draining low-pressure systems, as, for example, the separator in an exhaust-steam heating system. It consists essentially of two legs A and B, which may be close together or any distance apart, but the lengths of which must be sufficiently great to prevent pressure, acting through pipe I, from forcing the water out of B. C is a vent pipe extending to the air to prevent siphoning; O is the discharge for the condensed steam. In ordinary operation B is filled with water which is constantly overflowing, and A with steam and water, the total pressure in both legs being equal. The siphon trap is applicable for low pressure only, as it requires approximately 2.3 ft. of vertical space E for each lb. per sq. in. pressure in the pipe. Allowable head is represented by vertical distance N.

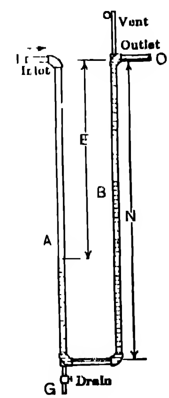


Fig. 514. Simple Siphon Trap.

Wherever possible, a trap should be located so that the condensation will flow into it by gravity. This will insure positive drainage. Sometimes, however, the coils, cylinders, or pipes to be drained are located in a pit or trench or lie on a basement floor where it is impossible to set the trap so as to receive the drips by gravity without placing it in an inaccessible position. With very low pressures this is often unavoidable, but with pressures of 5 lb. or more the trap may be placed above the point to be drained. If a trap is set in an exposed place, a drain should be provided at the lowest point to free the pipe of water when steam is shut off. A dirt catcher or strainer should be placed in the pipe leading to the trap to prevent scale, etc., from reaching the valve. All pockets and dead ends should be drained, and no condensation should be allowed to accumulate. High- and low-pressure drips should be kept separate. All tanks should have gage glasses.

Return Traps. — The traps previously illustrated, with the exception of the one shown in Fig. 508, are of the non-return or separating design and are intended primarily to discharge the condensate to any vessel having a pressure less than that of the actuating steam. In order to

return the condensate to the boiler or to remove it from a chamber under vacuum, it is necessary to add "equalizing" valves for the purpose of admitting high-pressure steam when the trap is ready to discharge and for relieving the pressure after the water has been disposed of. These valves are usually incorporated in the design of traps intended for this service. Figure 515 shows the basic principles of a return trap. The trap must be placed 3 ft. or more above the water line in the boiler so that the water may gravitate to the latter. When empty, the trap is vented to the atmosphere and is ready to receive the water from the return line A. If the pressure in the return pipe is not sufficient to lift the water into the trap, a pump or a non-return trap must be used to effect this result. Water flows into the return trap until it reaches the discharge point, when the equalizing valve closes the vent to the atmosphere and admits boiler steam into the body of the trap.

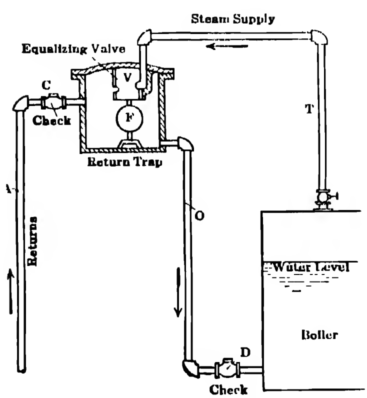


FIG. 515. Return-Trap Installation.

As soon as the head of the water in the discharge pipe O, plus that of the live steam in the trap, is greater than the pressure against check valve D, the water will gravitate into the boiler. At the end of discharge, the float or actuating mechanism shuts off the live steam and opens the vent to the atmosphere, and the trap is in position once more for receiving its supply from the return line. The check valve C prevents the water from being forced back to the return pipe while the trap is discharging.

When extracting condensate from a vacuum chamber, the trap must be placed so that the water will flow into it by gravity. When full, the actuating mechanism opens the steam valve for admission of live steam, which in turn forces the contents out of the trap through a discharge valve. When empty, the mechanism returns to the filling position, closes the steam and discharge valves and opens the vent to the vacuum chamber. The condensate then gravitates into the trap as before.

The **steam loop,** Fig. 516, while a practical means of returning high-pressure drips to the boiler in small plants, is little used because of the annoyance in starting up. It is of academic value, however, in showing how water may be returned to the boiler without the use of a trap, pump or injector.

In the figure the loop is returning the condensation from a steam separator to a boiler above the level of the separator. The apparatus is very

simple, consisting of one horizontal and two vertical lengths of plain pipe placed as indicated. Pipes *R* and *B* may be covered, but "horizontal" *A* is left uncovered, as its function is that of a condenser. The operation is as follows: Circulation is first started by opening stop valve *O* at the bottom of the drop leg until steam escapes. The valve is then closed and the steam in the horizontal *A* condenses and gravitates to the drop leg *B*. On account of the slight reduction in pressure in the horizontal,

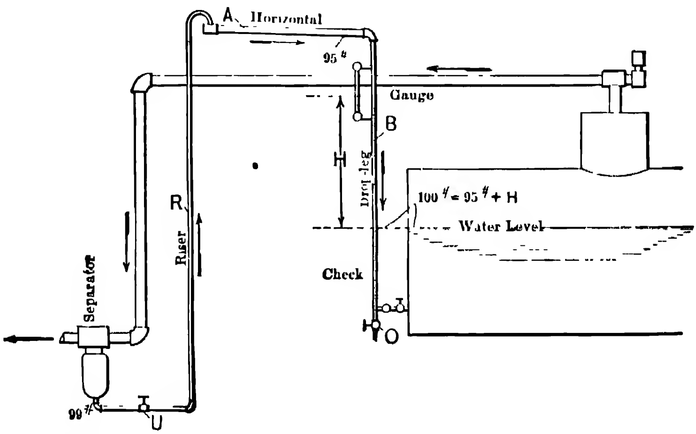


FIG. 516. General Arrangement of the Simple "Steam Loop."

a mixture of spray and steam flows from the separator chamber to the horizontal, and, condensing, gravitates to the drop leg. The column of water in the drop leg rises until its static head balances the difference of pressure in the riser *R* and the horizontal.

In other words, a decrease in pressure in the horizontal produces similar effects on the contents of the riser and drop leg but in a degree inversely proportional to their densities. Any further accumulation causes an equal amount to pass from the bottom of the column to the boiler, since the pressure in the boiler is then less than that at the bottom of the column; that is, the steam pressure on the top of the water column plus the hydrostatic head *H* is greater than the pressure in the boiler. Once started, the process is continuous and requires no further attention.

The **Holly loop** is an application of the steam loop to larger plants where there are many points requiring drainage. There are a number of the older plants employing this system of returning high-pressure drips to the boiler, but its use has been practically discontinued in the modern high-pressure central station. In the Holly system, all condensation is collected in a common receiver placed at the lowest point to be drained. The **horizontal** is a large cylindrical tank located at a considerable height

above the boiler room and connected with the feedwater heater through a reducing valve. A **riser** connects the receiving chamber with the upper tank, and a **drop leg** leads from the tank to the boiler drum below the lowest water level. In starting up, a valve in the bottom of the drop leg is opened to the atmosphere until there is a continuous flow of steam and water entrainment from receiver through this opening. The valve is then closed, and, by bleeding the upper tank through a reducing valve into the feedwater heater, a pressure is maintained in the upper tank or discharge chamber sufficiently below that of the steam in the receiver to permit of a continuous flow of steam and water spray from the receiver through the riser and into the discharge tank. The steam separates from the water and passes through the reducing valve into the heater, while the water collects in the bottom of the discharge tank and in the drop leg until the head of the steam and water is sufficient to overcome the resistance of the check valve in the boiler. The principles of operation are exactly the same as in the simple loop. The process is automatic and continuous so long as the plant is in operation.

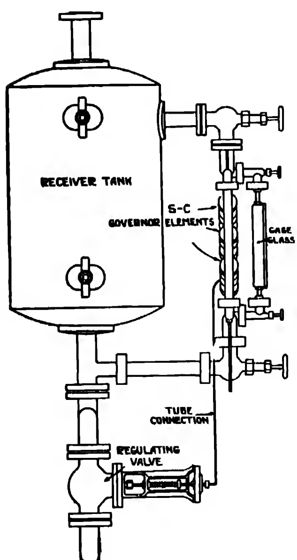


FIG. 517. "S-C" High-pressure Condensation Controller.

Figure 517 gives a general assembly of the **"S-C" high-pressure condensation controller** as installed in the Calumet, Crawford, Waukegan, and Philo stations. It is identical in principle to "S-C" feedwater regulator described in paragraph 292. The water of condensation gravitates to a suitable receiving tank and the fluctuations in level actuate a balanced valve in the discharge line through the agency of the "S-C" generator. Where the return lines are too long to permit of one central receiver, a small receiver may be placed near the condensation source since the apparatus is compact and requires but little space.

In the power plants of tall office buildings, the public sewers are often above the basement level, and it is necessary to remove all liquid wastes mechanically.

The **Shone pneumatic ejector,** which is in principle a float trap, has been found to serve this purpose effectually. This apparatus is placed in a pit in the basement floor into which all sewage, drips from engines, washings from boilers, and ground water gravitate, and are automatically discharged into the street sewer by means of compressed air.

Figure 518 gives a sectional view of a Shone ejector of ordinary construction. It consists essentially of a closed vessel furnished with inlet and discharge connections fitted with check valves, *A* and *B*, opening in opposite directions with regard to the ejector. Two cast-iron bells, *C* and *D*, are linked to each other, in reverse positions, and their rising and falling control the supply of compressed air through the agency of automatic valve *E*.

The bells are shown in their lowest position; the supply of compressed air is cut off from the ejector, and the inside of the vessel is open to the atmosphere. The sewage gravitating into the ejector raises the bell *C*, which in turn actuates the automatic valve *E*, thereby closing the connection between the inside of the ejector and the atmosphere and opening the connection with the compressed air. The air pressure expels the contents through the bell-mouthed opening at the bottom and the discharge valve *B* into the main sewer. Discharge continues until the level falls to such a point that the weight of the sewage retained in the bell *D* is sufficient to pull it down, thereby reversing the automatic valve. This cuts off the supply of compressed air and reduces the pressure to that of the atmosphere.

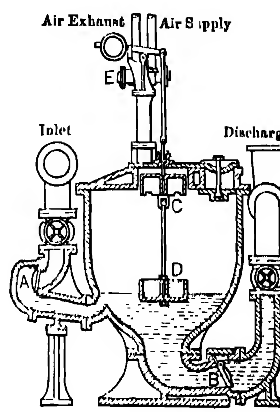


FIG. 518. Shone Ejector.

The positions of the bells are so adjusted that compressed air is not admitted until the ejector is full, and is not allowed to exhaust until emptied down to the discharge level; thus the ejector discharges a fixed quantity each time it operates.

Two ejectors, each of a capacity suitable for handling the average flow of tributary sewage and so arranged that they can work either independently or together, are usually installed at each ejector station.

The main sanitary sewer of the building usually discharges directly into the ejectors, the surface water, drips, etc., being collected in a neighboring sump. The latter is connected to the sanitary sewer through a trap or backwater valve.

Steam Traps — Their Selection, Installation and Upkeep: Power, July 11, 1922, p. 45.

Steam-trap Installation and Operation: Power, Oct. 9, 1923, p. 573.

Tests of Radiator Traps: Jour. A.S.H. & V.E., Mar., 1923, p. 79.

CHAPTER XVI

PIPING AND PIPE FITTINGS

298. General. — The main object in any steam power station is to use the least piping possible, consistent with the requirements of the heat balance, and at the same time to provide facilities for shutting down any portion of the different apparatus and piping without interfering with the service for which the station is intended. Simplicity and flexibility are of prime importance, but safety is the fundamental requirement and should not be sacrificed for economy.

While pipes, fittings, and valves have been pretty well standardized, there is no standard system of piping arrangement because of the diversified layout of each power station. Each plant is a separate problem and the piping must be arranged to conform to its specific requirements.

The engineer usually specifies the make, style, and size of valves, pipes, and fittings, furnishes drawings showing the location of the various boilers, pumps, etc., and indicates the approximate location of the pipe lines and fittings; but, as a rule, he leaves the exact details of construction and installation to the pipe contractor. Some idea of current practice in this respect may be gained from the piping specifications outlined in paragraph 374.

A detailed analysis of the design, installation, and operation of the various kinds of piping systems in power plants is beyond the scope of this book, and the reader is referred to Steam Power Plant Piping Systems, by Wm. L. Morris, McGraw Hill Co., Publishers, for extended study.

299. Materials for Pipes and Fittings. — An inspection of the curves in Fig. 98 will show that there is very little difference in the ultimate strength and yield point of the various metals used in the fabrication of pipes and fittings for temperatures between 70 and 450 deg. fahr. Above 450–500 deg. fahr., the ultimate strength and yield point begin to fall off, the rate of decrease varying widely with the character of the material. Therefore, for temperatures up to 450 deg. fahr. no attention need be paid to the temperature factor in proportioning the thickness of the various parts. In the modern central station involving temperatures of 750 deg. fahr. and in certain industrial plants where temperatures of 1000 deg. are not uncommon, the temperature element is an important factor in

the design because of the greatly reduced ultimate strength and yield point.

Wrought Steel. — The greater portion of the piping of the average steam power plant is of Bessemer or open-hearth steel made under the specifications of tube mills for this class of material. The tubes are lap-welded, hammer-welded, butt-welded, riveted, or seamless-drawn depending upon the size of pipe and the service for which they are intended. Wrought-steel pipe is cheaper than that manufactured from other materials and

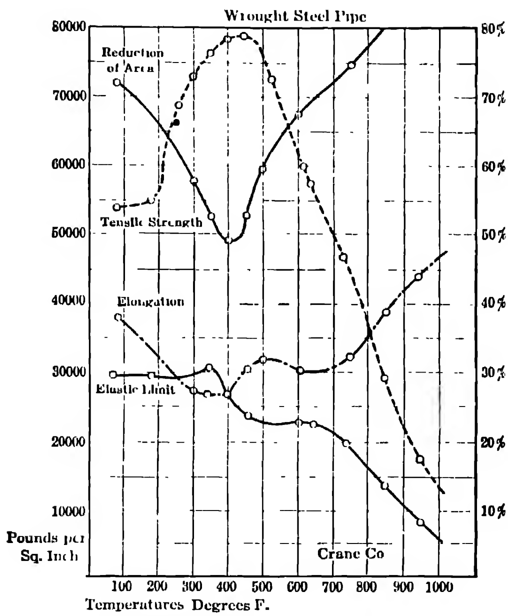


FIG. 519. Effect of High Temperature on the Physical Properties of Wrought-steel Pipe.

fulfills practically all requirements for general service. See Fig. 98 for influence of temperature on the ultimate strength, elastic limit, and elongation of wrought-steel pipe material. Pipe couplings, pressed-steel fittings, and certain grades of forged-steel flanges are also made from Bessemer or open-hearth steel.

A.S.T.M. Specifications for Welded and Seamless Steel Pipe: Am. Soc. Testing Materials, Standards, 1921, A53, p. 218.

Wrought Iron. — The term "wrought iron" in a commercial sense refers to Bessemer or open-hearth steel, and unless it is distinctly specified that genuine wrought iron is desired an order calling for wrought-iron

pipe will ordinarily be filled with the steel product. Genuine wrought iron is softer than steel and welds more readily, but its tensile strength is somewhat lower. It is commonly used for boiler tubes, and to a limited extent for water, gas, and steam pipes, but it is not much in evidence for high-pressure steam piping. Wrought-iron pipe has a longer life than steel under certain conditions:

A.S.T.M. Specifications for Seamless-steel and Wrought-iron Boiler Tubes for Stationary Boilers: Am. Soc. Testing Materials, Standards, 1921, A52, p. 210.

Cast Iron. — Companion flanges, valve bodies, manifold headers and special fittings are made of gray cast iron for steam pressures up to 250 lb. gage pressure and temperatures up to 450 deg. fahr. Cast iron is commonly used for underground water and gas service and in connection

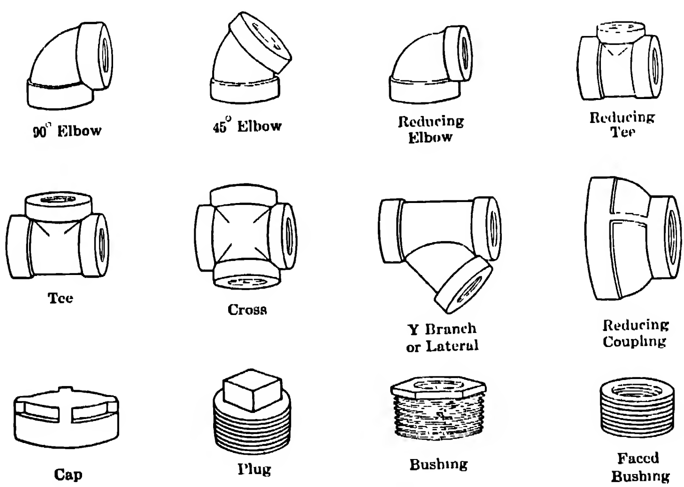


FIG. 520. Standard Screwed Cast-iron Fittings.

with certain industrial processes where the corrosion of wrought iron or steel would be excessive. The chief objections to cast iron for high-pressure steam are its weight, low tensile strength, and lack of ductility. While the tensile strength and yield point are little influenced by temperatures up to 1000 deg. fahr., it has been definitely proved that cast iron in actual service, when subjected to continued temperatures of approximately 450–600 deg. fahr., takes a permanent expansion set and does not return to its original volume when cooled. Cast iron is little used in the steam piping systems of the modern high-pressure central station.

Standard Specifications for Cast-iron Pipe and Special Fittings: Am. Soc. Testing Materials, Standards, 1921, A44, p. 336.

Cast Steel. — All fittings, valve bodies, and cast-metal manifold headers in the modern high-pressure high-temperature plant are made of cast steel, straight or alloyed. The ultimate strength and yield point is high even at temperatures of 900 deg. fahr. and there is but little permanent set after long-continued service at a temperature of 750 deg.

Malleable Iron and Semi-steel. — Small screwed fittings and various sizes of companion flanges are frequently made of malleable iron, which is midway between cast iron and cast steel in its composition and physical characteristics. Its tensile strength is about twice that of cast iron and it is not brittle and subject to breakage by a sudden blow. **Semi-steel** is a mixture of mild-steel scrap and pig iron with a small quantity of manganese or other special flux and it is used for larger-sized valves and fittings in which close grain and strength are needed but in which the temperature does not exceed 500 deg. fahr.

Miscellaneous Materials. — Copper, brass, bronze, and monel metals are the principal non-ferrous materials used for valves, fittings, or trimmings. Brass valves and fittings are used where salt water is to be handled, because of the resistance of this metal to the corrosive action of the water, but are not in evidence in steam lines, except for low pressures and temperatures, and then only for pipe sizes under 6 in. in diameter. **Monel metal,** because of its high tensile strength at high temperatures and high resistance against erosion and corrosion, is used almost exclusively for the internal trimmings of superheated-steam valves.

Copper pipes were in common use for steam for many years in marine service on account of their flexibility. To increase the bursting strength, pipes above 6 in. in diameter were generally wound with a close spiral of copper or composition wire. In recent years wrought-iron and steel pipe bends have practically superseded copper for flexible connections. As a rule the use of copper pipes should be avoided for high temperatures, on account of the rapid deterioration of the metal under temperature and stress variations. The cost is prohibitive for such purposes and this alone prevents it from being seriously considered in the manufacture of pipe. Copper expansion joints are occasionally used in low-pressure work.

Brass is little used in the construction of steam pipes except for certain industrial purposes, on account of its high cost. It withstands the corrosive action of air and moisture much better than iron or steel and is sometimes used in connecting the feed main with the boiler drum. Special alloys, **nickel steel, ferrosteel, malleable iron,** and the like have been used in the manufacture of pipes, and possess points of superiority over wrought iron and steel for some purposes, but the cost is prohibitive except for special applications.

Galvanized iron pipes and fittings resist ordinary corroding agencies

more readily than the bare metal, but the coating is rapidly destroyed by mine-water, tunnel gases and sea-water.

Lead-lined and **lead-covered** cast-iron, wrought-steel, and brass pipes are used where internal and external protection against acids, mine water, salt water, and the like is essential. **Tin-lined** brass pipes are also used for this purpose.

Protective Coatings for Pipe: Bul. No. 8 C, 1922, National Tube Co., Pittsburgh, Pa.
Properties of Metals at High Temperatures: Trans. A.S.M.E., Vol. 44, 1922, p. 1130; Power Plant Engrg., Aug. 1, 1924, p. 801; Power, June 24, 1924, p. 1020.

300. Size and Strength of Commercial Pipe. — Wrought-iron and mild-steel pipes are marketed in standard sizes. Those most commonly used in steam power plants are designated as

1. Merchant or standard pipe.
2. Full-weight pipe.
3. Extra heavy or extra strong.
4. Double extra heavy or double extra strong.
5. Large O.D. pipe.
6. Hydraulic.

Table 88 gives the dimensions of standard **full-weight** pipe, which is specified by the nominal inside diameter up to and including 12 in. and based on the Briggs standard. Wrought-steel pipes larger than 12 in. are designated by the actual outside diameter (O.D.) and are made in various weights as determined by the thickness of metal specified. Manufacturers specify that "full-weight" pipe may have a variation of 5 per cent above or 5 per cent below the nominal or table weights, but merchant pipe, which is the standard pipe of commerce, such as manufacturers and jobbers usually carry in stock, is almost invariably under the nominal weight. It varies somewhat among the different mills, but usually lies between 5 and 10 per cent under the table weight. The smaller sizes of merchant pipe, 1/8 in. to 3 in., are butt-welded and the larger sizes are lap-welded.

Extra-heavy and **double-extra-heavy** pipe have the same external diameter as the standard in order to accommodate pipe-thread standards, but are of greater thickness and hence have a smaller internal diameter. Taking the thickness of the standard pipe as 1, that of the extra heavy is approximately 1.4 and of the double extra heavy 2.8.

Hydraulic wrought-steel pipe is constructed only in 9, 10, 11 and 12-in. inside-diameter pipe sizes.

When it is desired to have the inner surface smooth, the pipe is "reamed and drifted."

TABLE 88

DIMENSIONS OF "STANDARD" WROUGHT PIPE

(National Tube Co.)

Size	Size	Diameters		Nominal Thickness	Circumference		Transverse Areas			Length of Pipe Per Sq. Ft. of		Length of Pipe Containing One Cubic Foot	Nominal Weight Per Ft.		Number of Threads per In. of Screw
		External	Approximate Internal		External	Internal	External	Internal	Metal	External Surface	Internal Surface		Plain Ends	Threaded and Coupled	
Inches	Millimeters	Inches	Inches	Inches	Inches	Inches	Sq. In.	Sq. In.	Sq. In.	Feet	Feet	Feet			
1/8	3	.405	.269	.068	1.272	.845	.129	.057	.072	9.431	14.199	2533.775	.244	.245	27
1/4	6	.540	.364	.088	1.696	1.144	.229	.104	.125	7.073	10.493	1383.789	.424	.425	18
3/8	10	.675	.493	.091	2.121	1.549	.358	.191	.167	5.658	7.747	754.360	.567	.568	18
1/2	13	.840	.622	.109	2.639	1.954	.554	.304	.250	4.547	6.141	473.906	.850	.852	14
3/4	19	1.050	.824	.113	3.299	2.589	.866	.533	.333	3.637	4.635	270.034	1.130	1.134	14
1	25	1.315	1.049	.133	4.131	3.296	1.358	.864	.494	2.904	3.641	166.618	1.678	1.684	11½
1¼	32	1.660	1.380	.140	5.215	4.335	2.164	1.495	.669	2.301	2.767	96.275	2.272	2.281	11½
1½	38	1.900	1.610	.145	5.969	5.058	2.835	2.036	.799	2.010	2.372	70.733	2.717	2.731	11½
2	50	2.375	2.067	.154	7.461	6.494	4.430	3.355	1.075	1.608	1.847	42.913	3.652	3.678	11½
2½	64	2.875	2.469	.203	9.032	7.757	6.492	4.788	1.704	1.328	1.547	30.077	5.793	5.819	8
3	76	3.500	3.068	.216	10.996	9.638	9.621	7.393	2.228	1.091	1.245	19.479	7.575	7.616	8
3½	90	4.000	3.548	.226	12.566	11.146	12.566	9.886	2.680	.954	1.076	14.565	9.109	9.202	8
4	100	4.500	4.026	.237	14.137	12.648	15.904	12.730	3.174	.848	.948	11.312	10.790	10.889	8
4½	113	5.000	4.506	.247	15.708	14.156	19.635	15.947	3.688	.763	.847	9.030	12.538	12.642	8
5	125	5.563	5.047	.258	17.477	15.856	24.306	20.006	4.300	.686	.756	7.198	14.617	14.810	8
6	150	6.625	6.065	.280	20.813	19.054	34.472	28.891	5.581	.576	.629	4.984	18.974	19.185	8
7	175	7.625	7.023	.301	23.955	22.063	45.664	38.738	6.926	.500	.543	3.717	23.544	23.769	8
8	200	8.625	8.071	.277	27.096	25.356	58.426	51.161	7.265	.442	.473	2.815	24.696	25.000	8
8	200	8.625	7.981	.322	27.096	25.073	58.426	50.027	8.399	.442	.478	2.878	28.554	28.809	8
9	225	9.625	8.941	.342	30.238	28.089	72.760	62.786	9.974	.396	.427	2.294	33.907	34.188	8
10	250	10.750	10.192	.279	33.772	32.019	90.763	81.585	9.178	.355	.374	1.765	31.201	32.000	8
10	250	10.750	10.136	.307	33.772	31.843	90.763	80.691	10.072	.355	.376	1.785	34.240	35.000	8
10	250	10.750	10.020	.365	33.772	31.479	90.763	78.855	11.908	.355	.381	1.826	40.483	41.132	8
11	275	11.750	11.000	.375	36.914	34.558	108.434	95.033	13.401	.325	.347	1.515	45.557	46.247	8
12	300	12.750	12.090	.330	40.055	37.982	127.676	114.800	12.876	.299	.315	1.254	43.773	45.000	8
12	300	12.750	12.000	.375	40.055	37.699	127.676	113.097	14.579	.299	.318	1.273	49.562	50.706	8

The Crane Company recommends the following rule for determining the proper weight or thickness of Bessemer or open-hearth wrought-steel steam piping to be used in power plants.

$$P = 2\,E(t - 0.08) \div Fd \tag{266}$$

in which

P = working pressure lb. per sq. in. gage,
E = elastic limit of the material at the temperature considered,
t = thickness of pipe, in.,
F = factor of safety = 4 for lap-welded and 5 for butt-welded,
d = outside diameter of pipe, in.

The values in Tables 89 to 90 are based upon equation (266), using 21,600 lb. per sq. in. as the elastic limit. This is the elastic limit, at 700 deg. fahr., which covers most commercial installations. (See Fig. 519.)

Tables 89 to 89*b* give the safe working pressure for full-weight, extra strong, hydraulic, and large O.D. pipes. Table 90 gives the proper pipe to be used for various pressures, as calculated from equation (266). These tables may be used for all temperatures up to and including 700 deg. fahr.

Standard full-weight wrought-steel pipes are marketed in random lengths varying from 12 to 20 ft. with plain or threaded ends. Extra strong and double extra strong pipe is shipped in random lengths with plain ends unless otherwise ordered.

Tables 88 to 90, inclusive, apply only to standard welded wrought-steel pipe. Standard welded wrought-iron pipe has a thicker wall than welded steel pipe and consequently a smaller internal diameter. Seamless tubes are seldom used for steam pipe lines but are much in evidence in the fabrication of steam boilers, superheaters, dry pipes, arch pipes and water grates. Riveted steel pipes are commonly used for hydraulic and low-pressure steam lines and to a limited extent for high-pressure steam lines. There are so many kinds of pipes on the market, designed for such a wide field of application, that no attempt will be made even to enumerate them, and the reader is referred to the various publications issued by the National Tube Co., Frick Building, Pittsburgh, Pa., for descriptive details.

Estimated Fabricated Pipe Costs: Power, Mar. 16, 1926, p. 414; Mar. 23, 1926, p. 443.

TABLE 89

ALLOWABLE WORKING PRESSURE FOR FULL-WEIGHT AND EXTRA HEAVY WROUGHT-STEEL PIPE

(Crane Co.)

Full-weight							Extra Heavy						
Size	Weight	Working Pressure		Size	Weight	Working Pressure	Size	Weight	Working Pressure		Size	Weight	Working Pressure
		B.	L.			L.			B.	L.			L.
½	0.85	298	..	5	14.62	345	½	1.09	690		5	20.78	573
¾	1.13	271		6	18.97	326	¾	1.47	609		6	28.57	574
1	1.68	348	.	7	23.54	313	1	2.17	650	..	7	38.05	595
1¼	2.27	312		8	24.70	246	1¼	3.00	578	..	8	43.39	526
1½	2.72	296		8	28.55	303	1½	3.63	546		9	48.73	471
2	3.65	269	336	9	33.91	291	2	5.02	501	627	10	54.74	421
2½	5.79	370	462	10	31.20	200	2½	7.66	588	736	11	60.08	386
3	7.58	336	420	10	34.24	228	3	10.25	513	679	12	65.42	386
3½	9.11		394	10	40.48	286	3½	12.51		643			
4	10.79	.	377	11	45.56	271	4	14.98	.	617			
4½	12.54		360	12	43.77	212	4½	17.61		594			
					49.56	250							

Weight in Lb per Lineal Ft
B Butt Welded

Pressure, Lb per Sq In Gage
L Lap Welded.

TABLE 89a

ALLOWABLE WORKING PRESSURES FOR LARGE O. D. WROUGHT STEEL PIPE

(Crane Co.)

O.D In.	Thickness, In.												
	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$
14	131	180	227	276	324	372	420	517	613	710	806	903	1000
15	122	168	212	258	302	348	392	482	572	662	753	843	933
16	115	157	199	242	284	326	368	452	337	621	706	790	875
17	108	148	187	227	267	307	346	426	505	585	665	743	823
18	102	140	177	215	252	290	327	402	477	552	627	702	777
20	.	126	159	193	227	261	294	362	430	497	565	632	700
21	.	120	152	184	216	248	280	341	409	473	538	602	666
22		114	145	175	206	237	268	329	390	452	513	575	636
24	..		133	161	189	217	245	302	358	414	470	526	582
26	.	.	122	149	174	200	226	278	.	..		..	...
28		.	...	138	162	186	210	258	..	.	...	..	...
30		.	...	129	151	174	196	241	..	.	.		...

TABLE 89b

ALLOWABLE WORKING PRESSURES FOR HYDRAULIC WROUGHT-STEEL PIPE

(Crane Co.)

Size	External Diameter, In	Thickness			
		5/8	3/4	7/8	1
9	9 5/8	611	751	892	1030
10	10 3/4	547	673	798	924
11	11 3/4	502	616	731	846
12	12 3/4	462	567	673	780

TABLE 90

WEIGHT OR THICKNESS OF PIPE TO BE USED ON VARIOUS STEAM PRESSURES, LB. PER LINEAR FT.

(Maximum Temperature 700 Deg. Fahr.)

(Crane Co.)

Size	Pressures									
	150	200	250	300	350	400	450	500	550	600
1/2	F	F	F	F	X	X	X	X	X	X
1	F	F	F	F	X	X	X	X	X	X
1 1/4	F	F	F	F	X	X	X	X	X	X
2	F	F	F	F	X	X	X	†X	X	†X
2 1/2	F	F	F	F	*X	X	X	X	X	†X
3	F	F	F	F	*X	X	X	†X	X	†X
3 1/2	F	F	F	F	*X	X	X	X	X	X
4	F	F	F	F	*X	X	X	X	X	X
4 1/2	F	F	F	F	*X	X	X	X	X	X
5	F	F	F	F	X	X	X	X	X	X
6	F	F	F	F	X	X	X	X	X	X
7	F	F	F	F	X	X	X	X	X	X
8	24.7	24.7	28.5	28.5	X	X	X	X	XX	XX
9	33.9	33.9	33.9	X	X	X	X	5/8H	5/8H	5/8H
10	31.2	34.2	40.5	X	X	X	5/8H	5/8H	3/4H	3/4H
11	45.5	45.5	45.5	X	X	5/8H	5/8H	5/8H	3/4H	3/4H
12	43.8	43.8	43.8	49.5	X	5/8H	5/8H	3/4H	3/4H	7/8H
14OD	5/16	3/8	7/16	1/2	9/16	5/8	3/4	3/4	7/8	7/8
15OD	5/16	3/8	7/16	1/2	5/8	3/4	3/4	7/8	7/8	1
16OD	5/16	3/8	7/16	9/16	5/8	3/4	3/4	7/8	1	1
17OD	3/8	7/16	1/2	9/16	3/4	3/4	7/8	7/8	1	1 1/8
18OD	3/8	7/16	1/2	5/8	3/4	3/4	7/8	1	1	1 1/8
20OD	3/8	1/2	9/16	3/4	3/4	7/8	1	1 1/8	1 1/8	1 1/4
22OD	7/16	1/2	5/8	3/4	7/8	1	1	1 1/8	1 1/4	1 3/8
24OD	7/16	9/16	3/4	3/4	7/8	1	1 1/8	1 1/8	1 3/8	1 1/2

F = Full-weight, X = Extra strong, XX = Double extra strong, * = Full-weight lap-welded may be used, † = Should be lap-welded, H = Hydraulic pipe.

301. Pipe Connections. — In erecting a pipe line, the different lengths of pipe are joined to each other and to the various fittings by **threaded, bolted, calked,** and **welded** joints. These joints may be applied to any class of pipe, the particular connection best suited for the purpose depending upon the size and material of the pipe and the service for which it is intended. Pipes with threaded ends may be joined together by means of (1) **wrought couplings,** Fig. 521; (2) **nut unions,** Fig. 522, which are

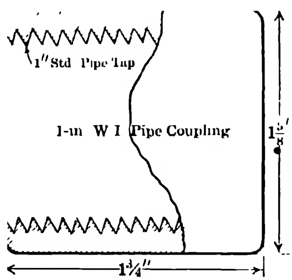


FIG. 521. Wrought Coupling.

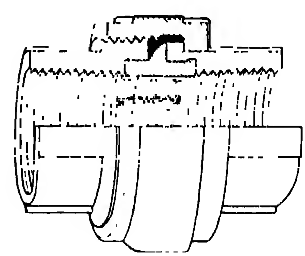


FIG. 522. Malleable Nut Union.

intended for pipe sizes of 3 in. and under, and (3) **flanged unions,** Fig. 525, *A*, *B*, *C*, etc. Wrought couplings are often used where runs of pipes are longer than lengths available or where cut pipe is used, and the nut and flanged unions where the last fitting is made up and where the joint may be taken apart. Bolted couplings are used invariably on pipes and fittings with flanged ends. **Bell-and-spigot** joints, Fig. 523, calked with lead, Gannister packing, or cement, are used mostly on water and low-pressure gas lines. This type of joint is so designed as to permit slight deviations from the regular alignment of the line without the use of special fittings and bends. This feature is particularly desirable in trenches which follow the contour of rolling or hilly country or which have not been leveled on the bottom. There are so many types and designs of joints on the market that it is impossible to cover the subject in a work of this nature, and the reader is referred for extended study to publications issued by the National Tube Company, Frick Building, Pittsburgh, Pa.; Crane Company, Chicago, Ill.; Grinnell Company, Chicago, Ill.; and other pipe manufacturers. Only a few of the joints commonly used in steam power plant service are briefly described in this chapter.

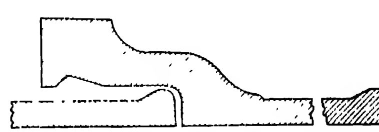


FIG. 523. Bell-and-Spigot Joint for C. I. Pipe (A. W. W. Standard).

All pipe connections and fittings should be designed in accordance with

established standards. The " American " standards, as recommended by the A.S.M.E., are universally used in this country for wrought-metal pipe connections and fittings, but there is no single standard for cast iron. The American Waterworks Specifications are used for cast-iron water pipe, and the American Gas Institute Standard for cast-iron gas pipe.

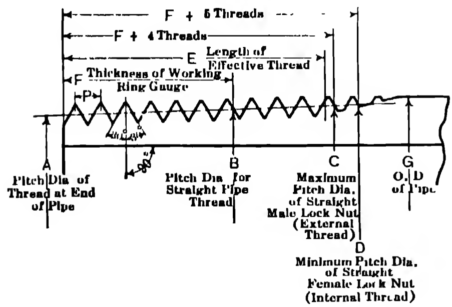


FIG 524 "American Standard" Pipe Thread

Screwed Connections. — The details of the national **American Standard thread** are shown in Fig. 524. (*Trans. A.S.M.E.*, Vol. 41, 1919, p. 1067.) The end of the pipe is tapered 1 to 16 measured on the diameter, the angle of the thread being 60 deg. with the axial plane. The crest and the root are truncated an amount equal to 0.033 of the pitch (P). The depth of the thread, therefore, is 0.8 P. The length of the thread and other dimensions are determined from the following rules:

$$A = G - (0.05\,G + 1.1)\,P \tag{267}$$

$$B = A + 0.0625\,F \tag{267a}$$

$$E = P\,(0.8\,G + 6.8) \tag{267b}$$

in which

A = pitch diameter at the end of the pipe, in.
G = outside diameter of the pipe, in.
P = pitch of thread, in.
B = pitch diameter at the gaging notch.
F = normal engagement by hand between male and female threads.
E = length of effective thread.

The maximum allowable variation in the commercial product is one turn plus or minus from the gaging notch when using working gages.

The length of thread screwed into valves or fittings in order to make a tight joint is given in Table 91.

When properly made, a screwed joint will hold against any pressure consistent with the strength of the pipe. The threads, however, are often poorly cut and the parts screwed together improperly cleaned and lubricated, thus causing leakage between the threads.

TABLE 91

LENGTH OF THREAD ON PIPE

(All Dimensions in Inches)
American Standard

Size of Pipe	Length of Thread	Size of Pipe	Length of Thread	Size of Pipe	Length of Thread
$\frac{1}{8}$	$\frac{1}{4}$	$1\frac{1}{2}$	$\frac{5}{8}$	5	$1\frac{3}{16}$
$\frac{1}{4}$	$\frac{3}{8}$	2	$\frac{11}{16}$	6	$1\frac{1}{4}$
$\frac{3}{8}$	$\frac{3}{8}$	$2\frac{1}{2}$	$\frac{11}{16}$	7	$1\frac{1}{4}$
$\frac{1}{2}$	$\frac{1}{2}$	3	1	8	$1\frac{5}{16}$
$\frac{3}{4}$	$\frac{1}{2}$	$3\frac{1}{2}$	$1\frac{1}{16}$	9	$1\frac{3}{8}$
1	$\frac{9}{16}$	4	$1\frac{1}{16}$	10	$1\frac{1}{2}$
$1\frac{1}{4}$	$\frac{5}{8}$	$4\frac{1}{2}$	$1\frac{1}{8}$	12	$1\frac{5}{8}$

Above dimensions do not allow for variation in tapping or threading

Flanged Connections. — Figure 525 illustrates some of the more commonly known methods of fitting flanges to the ends of wrought-metal pipes.

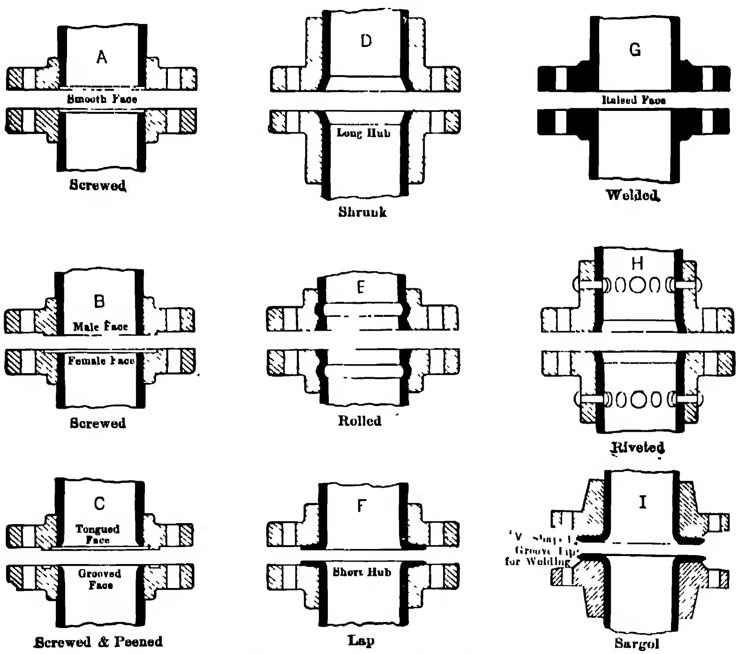


FIG. 525. Types of Pipe Flanges.

In *A* to *C*, Fig. 525, the pipes are screwed into cast-iron or forged-steel flanges and the two faces, with metallic or composition gasket between,

are drawn together by bolts. *A* illustrates the most common and inexpensive of flanged joints, and requires no special tools and can be made up at the place of erection. It gives satisfactory results for pressures of 100 lb. or less, but for higher pressures leakage is apt to take place between the threads. The flanges are sometimes made with a long thread and a recess which can be calked with soft metal. A similar joint is made with the pipe screwed beyond the face of the flange and the two faced off together, either plain or as shown in *B*, which is known as a **male** and **female,** or **hydraulic,** joint. This method forms a very reliable joint, since the ends of the pipe bear on the gasket, and the gasket is prevented from being blown out. An objection lies in the difficulty of opening the line to remove the gasket or replace a fitting. *C* is a modification known as the **tongued** and **grooved** joint, which uses an extremely narrow gasket. Such flanges may be subjected to severe strains when the bolts are drawn up, owing to the small area of contact. Metal gaskets are recommended, since soft material is apt to be squeezed out. In *C* the ends of the pipe are **peened,** which is an improvement over the simple screwed joint. *D* illustrates a **shrunk** joint. The flanges are bored for a shrink fit and forced over the pipe when at a red heat. After cooling, the end is beaded over into a recess on the face of the flange and a light cut taken from both. *H* shows a modification in which the hub is riveted to the pipe. *E* illustrates a joint constructed by rolling the pipe into a corrugation in the flange. The end of the pipe is then faced off flush.

A very successful commercial joint is illustrated by *F* and is known as the **lap** or **Van Stone** type. The pipe is expanded as indicated, and a light cut is then taken from the flared ends to insure a tight joint. The flanges are loose and permit of considerable flexibility in shifting them through various angles. Soft steel, copper, monel metal, aluminum and various types of prepared asbestos gaskets are used with this joint. For very high steam pressures and temperatures up to 750 deg. fahr., the pure, soft-annealed, aluminum gasket appears to give satisfaction.

Pipes with flanges **welded** on the end, as in *G*, are used in a number of central stations for high-pressure and high-temperature work. The faces are ordinarily raised 1/32 to 1/16 in. inside the bolt holes and ground to a steam-tight fit, so that thick gaskets are unnecessary. Tongue-and-groove faces are also used for high-pressure steam service.

For ordinary pressures and temperatures, any of the joints when well made will prove satisfactory. For extremely high pressures and temperatures, Van Stone joints with full thickness of pipe at the weld, or hammer-welded joints, are standard practice. In the different designs of Van Stone joints in use, the flanges themselves are proportioned in a variety of ways but the basic principles involved are the same.

Figure 526 shows the Crane Company design for pressures up to 800 lb. and temperatures up to 750 deg. fahr. It will be noted that the full thickness of metal has been maintained at the end of the pipe. The **Sargol joint,** Fig. 525 *I*, which is used in a number of stations for pressures of 400 to 500 lb. gage, differs from the usual design of Van Stone joint in that no gasket is used and the joint is sealed by welding the edges of the pipe lap.

Details of Sargol Joints: Report of Prime Movers Committee, N.E.L.A., T5-21, 1921, p. 137; Power Plant Engrg., Aug. 15, 1923, p. 819.

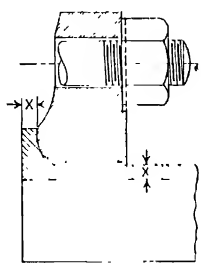


FIG. 526. Crane "Full Thickness Lap."

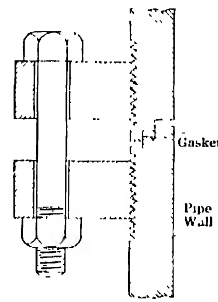


FIG. 527. Screwed Flange Coupling for High Pressures.

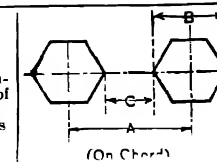
Figure 527 shows a flanged joint suitable for small pipes in which a very high pressure is to be carried. This is only applicable to double extra heavy lap-welded pipe or heavy seamless-drawn tubing.

Experiments conducted by the M. W. Kellogg Co. show that the ratio of gasket to internal hydraulic pressure necessary to keep a joint tight should not be less than 12 to 1. In order to maintain a tight joint with internal pressures of 400 lb. per sq. in. or more, alloy-steel bolts must be used in place of the customary mild-steel bolts.

All flanged connections for wrought pipe should be proportioned in accordance with the **American Engineering Standards** which have been adopted by all American and Canadian manufacturers. Dimensions for the American standard for various types of standard, low-pressure and extra heavy cast-iron fittings and connections for pressures up to 250 lb. are given in Tables 92–3. American standards for flanges and fittings at 750 deg. fahr. for pressures of 250, 400, 600, 900, 1350, 2000, and 3200 lb. may be obtained from the Secretary of the American Society of Mechanical Engineers, 29 W. 39th St., New York City. The Crane Co. recommendations for flanges, bolting and flange facing for 400–600 lb. steam working pressures may be found in the Report of Prime Movers Committee, N.E.L.A., 1923, Part A, p. 33.

TABLE 92

AMERICAN FLANGE STANDARD



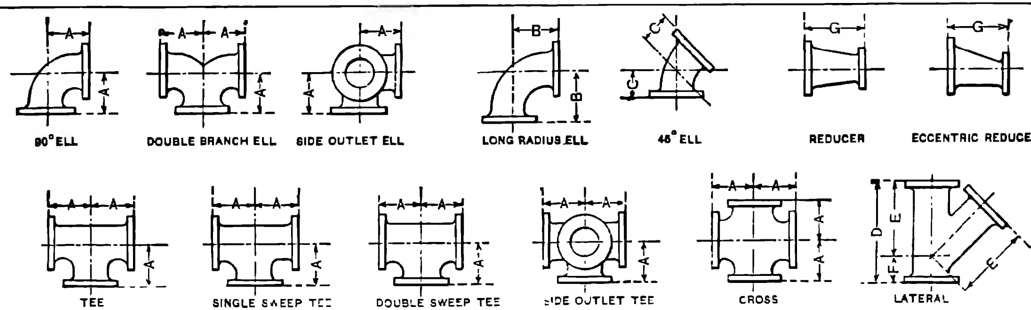
Diameter of Pipe	Thickness of Pipe Cast Iron	Minimum Thickness (Fractions of an Inch)	Stress on Pipe per Sq In	Diameter of Flange	Thickness of Flange	Width of Flange Face	Diameter of Bolt Circle	No of Bolts	Diameter of Bolts	Effective Area	Stress per Sq In. on Bolt Metal	Diameter of Bolt Holes	A	B	C
Standard Weight Flanges															
1	0 43	$\frac{7}{16}$	143	4	$\frac{7}{16}$	$1\frac{1}{2}$	3	4	$\frac{7}{16}$	0 093	264	$\frac{9}{16}$	2 12	0 91	1 21
$1\frac{1}{4}$	0.44	$\frac{7}{16}$	178	$4\frac{1}{2}$	$\frac{1}{2}$	$1\frac{5}{8}$	$3\frac{3}{8}$	4	$\frac{7}{16}$	0 093	412	$\frac{9}{16}$	2 38	0 91	1.47
$1\frac{1}{2}$	0 45	$\frac{7}{16}$	214	5	$\frac{9}{16}$	$1\frac{3}{4}$	$3\frac{7}{8}$	4	$\frac{1}{2}$	0 126	438	$\frac{5}{8}$	2 73	1 00	1.73
2	0 46	$\frac{7}{16}$	286	6	$\frac{5}{8}$	2	$4\frac{3}{4}$	4	$\frac{5}{8}$	0.202	486	$\frac{3}{4}$	3 35	1.21	2.14
$2\frac{1}{2}$	0 48	$\frac{7}{16}$	357	7	$\frac{11}{16}$	$2\frac{1}{4}$	$5\frac{1}{2}$	4	$\frac{5}{8}$	0 202	750	$\frac{3}{4}$	3 88	1 21	2 67
3	0 50	$\frac{7}{16}$	428	$7\frac{1}{2}$	$\frac{3}{4}$	$2\frac{1}{4}$	6	4	$\frac{5}{8}$	0 202	1093	$\frac{3}{4}$	4 23	1 21	3.02
$3\frac{1}{2}$	0 52	$\frac{7}{16}$	500	$8\frac{1}{2}$	$\frac{13}{16}$	$2\frac{1}{2}$	7	4	$\frac{5}{8}$	0 202	1488	$\frac{3}{4}$	4 94	1 21	3 73
4	0 53	$\frac{1}{2}$	500	9	$\frac{15}{16}$	$2\frac{1}{2}$	$7\frac{1}{2}$	8	$\frac{5}{8}$	0 202	972	$\frac{3}{4}$	2 87	1 21	1 56
$4\frac{1}{2}$	0.55	$\frac{1}{2}$	562	$9\frac{1}{4}$	$\frac{15}{16}$	$2\frac{3}{8}$	$7\frac{3}{4}$	8	$\frac{3}{4}$	0 302	823	$\frac{7}{8}$	2 96	1.44	1.52
5	0 56	$\frac{1}{2}$	625	10	$\frac{15}{16}$	$2\frac{1}{2}$	$8\frac{1}{2}$	8	$\frac{3}{4}$	0 302	1016	$\frac{7}{8}$	3 25	1 44	1.81
6	0 60	$\frac{9}{16}$	667	11	1	$2\frac{1}{2}$	$9\frac{1}{2}$	8	$\frac{3}{4}$	0 302	1463	$\frac{7}{8}$	3 63	1 44	2 19
7	0 63	$\frac{5}{8}$	700	$12\frac{1}{2}$	$1\frac{1}{16}$	$2\frac{3}{4}$	$10\frac{3}{4}$	8	$\frac{3}{4}$	0 302	1991	$\frac{7}{8}$	4.11	1 44	2 67
8	0 66	$\frac{5}{8}$	800	$13\frac{1}{2}$	$1\frac{1}{8}$	$2\frac{3}{4}$	$11\frac{3}{4}$	8	$\frac{3}{4}$	0 302	2600	$\frac{7}{8}$	4 50	1.44	3.06
9	0 70	$\frac{11}{16}$	818	15	$1\frac{1}{8}$	3	$13\frac{1}{4}$	12	$\frac{3}{4}$	0 302	2194	$\frac{7}{8}$	3 43	1 44	1.99
10	0 73	$\frac{3}{4}$	833	16	$1\frac{3}{16}$	3	$14\frac{1}{4}$	12	$\frac{7}{8}$	0.420	1948	1	3 69	1 66	2.03
12	0 80	$\frac{13}{16}$	923	19	$1\frac{1}{4}$	$3\frac{1}{2}$	17	12	$\frac{7}{8}$	0 420	2805	1	4 40	1 66	2.74
14	0 86	$\frac{7}{8}$	1000	21	$1\frac{3}{8}$	$3\frac{1}{2}$	$18\frac{3}{4}$	12	1	0 550	2915	$1\frac{1}{8}$	4 86	1 88	2 98
15	0 90	$\frac{7}{8}$	1072	$22\frac{1}{4}$	$1\frac{3}{8}$	$3\frac{5}{8}$	20	16	1	0 550	2510	$1\frac{1}{4}$	3 90	1 88	2.02
16	0 93	1	1000	$23\frac{1}{2}$	$1\frac{7}{16}$	$3\frac{3}{4}$	$21\frac{1}{4}$	16	1	0 550	2856	$1\frac{1}{8}$	4 14	1 88	2.26
18	1.00	$1\frac{1}{16}$	1059	25	$1\frac{9}{16}$	$3\frac{1}{2}$	$22\frac{3}{4}$	16	$1\frac{1}{8}$	0 694	2865	$1\frac{1}{4}$	4 44	2 09	2.35
20	1.07	$1\frac{1}{8}$	1111	$27\frac{1}{2}$	$1\frac{11}{16}$	$3\frac{3}{4}$	25	20	$1\frac{1}{8}$	0 694	2829	$1\frac{1}{4}$	3 91	2.09	1.82
22	1.13	$1\frac{3}{16}$	1158	$29\frac{1}{2}$	$1\frac{13}{16}$	$3\frac{3}{4}$	$27\frac{1}{4}$	20	$1\frac{1}{4}$	0 893	2660	$1\frac{3}{8}$	4 26	2 31	1 95
24	1.20	$1\frac{1}{4}$	1200	32	$1\frac{7}{8}$	4	$29\frac{1}{2}$	20	$1\frac{1}{4}$	0 893	3166	$1\frac{3}{8}$	4 62	2.31	2.31

Extra Heavy Flanges

1	0.45	1/2	250	4½	11/16	1¾	3¼	4	1/2	0.126	389	5/8	2.29	1.00	1.29
1¼	0.47	1/2	312	5	3/4	1⅞	3¾	4	1/2	0.126	609	5/8	2.65	1.00	1.65
1½	0.49	1/2	375	6	13/16	2¼	4½	4	5/8	0.202	547	3/4	3.17	1.21	1.96
2	0.51	1/2	500	6½	7/8	2¼	5	4	5/8	0.202	972	3/4	3.53	1.21	2.32
2½	0.53	9/16	555	7½	1	2½	5⅛	4	3/4	0.302	1016	7/8	4.15	1.44	2.71
3	0.56	9/16	667	8¼	1⅛	2⅝	6⅝	8	3/4	0.302	731	7/8	2.53	1.44	1.09
3½	0.59	9/16	778	9	1 3/16	2¾	7¼	8	3/4	0.302	995	7/8	2.77	1.44	1.33
4	0.61	5/8	800	10	1¼	3	7⅝	8	3/4	0.302	1300	7/8	3.01	1.44	1.57
4½	0.64	5/8	900	10½	1 5/16	3	8½	8	3/4	0.302	1646	7/8	3.25	1.44	1.81
5	0.67	11/16	909	11	1⅜	3	9¼	8	3/4	0.302	2032	7/8	3.53	1.44	2.09
6	0.72	3/4	1000	12½	1 7/16	3¼	10⅝	12	3/4	0.302	1950	7/8	2.75	1.44	1.31
7	0.78	13/16	1077	14	1½	3½	11⅞	12	7/8	0.420	1909	1	3.07	1.66	1.41
8	0.83	13/16	1230	15	1⅝	3½	13	12	7/8	0.420	2493	1	3.36	1.66	1.70
9	0.89	7/8	1285	16¼	1¾	3⅝	14	12	1	0.550	2410	1¼	3.62	1.88	1.74
10	0.94	15/16	1333	17½	1⅞	3¾	15¼	16	1	0.550	2231	1⅛	2.97	1.88	1.09
12	1.05	1	1500	20½	2	4¼	17¾	16	1⅛	0.694	2546	1¼	3.46	2.09	1.37
14	1.16	1⅛	1555	23	2⅛	4½	20¼	20	1⅛	0.694	2773	1¼	3.17	2.09	1.08
15	1.21	1 3/16	1579	24½	2 3/16	4¾	21½	20	1¼	0.893	2473	1⅜	3.36	2.31	1.05
16	1.27	1¼	1600	25½	2¼	4¾	22½	20	1¼	0.893	2814	1⅜	3.52	2.31	1.21
18	1.37	1⅜	1636	28	2⅜	5	24¾	24	1¼	0.893	2968	1⅜	3.23	2.31	0.92
20	1.48	1½	1666	30½	2½	5¼	27	24	1⅜	1.057	3096	1½	3.52	2.53	0.99
22	1.59	1 9/16	1760	33	2⅝	5¼	29¼	24	1½	1.295	3058	1⅝	3.81	2.75	1.06
24	1.70	1⅝	1846	36	2¾	5¾	32	24	1⅝	1.515	3110	1¾	4.18	2.96	1.22
26	1.81	1 13/16	1793	38¼	2 13/16	6⅛	34½	28	1⅝	1.515	3126	1¾	3.86	2.96	0.90
28	1.91	1⅞	1866	40¾	2 15/16	6⅜	37	28	1⅝	1.515	3629	1¾	4.14	2.96	1.18

TABLE 93

AMERICAN FLANGE STANDARD



Standard Flanged Fittings. — Straight Sizes

	Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15	16	18	20	22	24
A–A	Face to face	7	7½	8	9	10	11	12	13	14	15	16	17	18	20	22	24	28	29	30	33	36	40	44
A	Center to face	3½	3¾	4	4½	5	5½	6	6½	7	7½	8	8½	9	10	11	12	14	14½	15	16½	18	20	22
B	Center to face of long radius ells	5	5½	6	6½	7	7¾	8½	9	9½	10¼	11½	12¾	14	15¼	16½	19	21½	22¾	24	26½	29	31½	34
C	Center to face of 45-deg. ells	1¾	2	2¼	2½	3	3	3½	4	4	4½	5	5½	5½	6	6½	7½	7½	8	8	8½	9½	10	11
D	Face to face laterals	7½	8	9	10½	12	13	14½	15	15½	17	18	20½	22	24	25½	30	33	34½	36½	39	43	46	49½
E	Center to face	5¾	6¼	7	8	9½	10	11½	12	12½	13½	14½	16½	17½	19½	20½	24½	27	28½	30	32	35	37½	40½
F	Center to face	1¾	1¾	2	2½	2½	3	3	3	3	3½	3½	4	4½	4½	5	5½	6	6	6½	7	8	8½	9
G	Face to face reducer	.	.	. .	.	.	6	6½	7	7½	8	9	10	11	11½	12	14	16	17	18	19	20	22	24
	Diameter of flange	4	4½	5	6	7	7½	8½	9	9¼	10	11	12½	13½	15	16	19	21	22¼	23½	25	27½	29½	32
	Thickness of flange	7/16	1/2	9/16	5/8	11/16	3/4	13/16	15/16	15/16	15/16	1	1 1/16	1⅛	1⅛	1 3/16	1¼	1⅜	1⅜	1 7/16	1 9/16	1 11/16	1 13/16	1⅞
	Diameter of bolt circle	3	3⅜	3⅞	4¾	5½	6	7	7½	7¾	8½	9½	10¾	11¾	13¼	14¼	17	18¾	20	21¼	22¾	25	27¼	29½
	No. of bolts	4	4	4	4	4	4	4	8	8	8	8	8	8	12	12	12	12	16	16	16	20	20	20
	Diameter of bolts	7/16	7/16	1/2	5/8	5/8	5/8	5/8	5/8	3/4	3/4	3/4	3/4	3/4	3/4	7/8	7/8	1	1	1	1¼	1⅛	1¼	1¼
	Minimum metal thickness of body	7/16	7/16	7/16	7/16	7/16	7/16	7/16	1/2	1/2	1/2	9/16	5/8	5/8	11/16	3/4	13/16	7/8	7/8	1	1 1/16	1⅛	1 3/16	1¼

Extra Heavy Flanged Fittings. — Straight Sizes

	Size	1	1¼	1½	2	2½	3	3½	4	4½	5	6	7	8	9	10	12	14	15	16	18	20	22	24
A-A	Face to face	8	8½	9	10	11	12	13	14	15	16	17	18	20	21	23	26	30	31	33	36	39	41	45
A	Center to face	4	4¼	4½	5	5½	6	6½	7	7½	8	8½	9	10	10½	11½	13	15	15½	16½	18	19½	20½	22½
B	Center to face of long radius ells	5	5½	6	6½	7	7¾	8½	9	9½	10¼	11½	12¾	14	15¼	16½	19	21½	22¾	24	26½	29	31½	34
C	Center to face of 45-deg. ells	2	2½	2¾	3	3½	3½	4	4½	4½	5	5½	6	6	6½	7	8	8½	9	9½	10	10½	11	12
D	Face to face, laterals	8½	9½	11	11½	13	14	15½	16½	18	18½	21½	23½	25½	27½	29½	33½	37½	39½	42	45½	49	53	57½
E	Center to face, laterals	6½	7¼	8½	9	10½	11	12½	13½	14½	15	17½	19	20½	22½	24	27½	31	33	34½	37½	40½	43½	47½
F	Center to face, laterals	2	2¼	2½	2½	2½	3	3	3	3½	3½	4	4½	5	5	5½	6	6½	6½	7½	8	8½	9½	10
G	Face to face, reducer						6	6½	7	7½	8	9	10	11	11½	12	14	16	17	18	19	20	22	24
	Diameter of flange	4½	5	6	6½	7½	8¼	9	10	10½	11	12½	14	15	16¼	17½	20½	23	24½	25½	28	30½	33	36
	Thickness of flange	11/16	¾	13/16	⅞	1	1⅛	1 3/16	1¼	1 5/16	1⅜	1 7/16	1½	1⅝	1¾	1⅞	2	2⅛	2 3/16	2¼	2⅜	2½	2⅝	2¾
	Diameter of bolt circle	3¼	3¾	4½	5	5⅞	6⅝	7¼	7⅞	8½	9¼	10⅝	11⅞	13	14	15¼	17¾	20¼	21½	22½	24¾	27	29¼	32
	No. of bolts	4	4	4	4	4	8	8	8	8	8	12	12	12	12	16	16	20	20	20	24	24	24	24
	Diameter of bolts	½	½	⅝	⅜	¾	¾	¾	¾	¾	¾	¾	⅞	⅞	1	1	1⅛	1⅛	1¼	1¼	1¼	1⅜	1½	1⅝
	Minimum metal thickness of body	½	½	½	½	9/16	9/16	9/16	⅝	⅝	11/16	¾	13/16	13/16	⅞	15/16	1	1⅛	1 3/16	1¼	1⅜	1½	1 9/16	1⅝

NOTES. — Figures given are for center to face and for face to face finished dimensions.
Where necessary manufacturers will make suitable allowances in patterns before casting.

Welded Joints. — Autogenously welded (oxy-acetylene) butt joints are used to a considerable extent in long pipe lines subjected to moderate temperature ranges but have not always proved reliable for high-pressure steam. Several types of welds in which the joints are reinforced with sleeves welded on over butt joints have proved eminently satisfactory under most severe conditions of use, but they are not much in evidence in the modern steam power plant. The majority of specifications call for Van Stone or Sargol joints for high-pressure steam work.

Outlets for branch connections are frequently welded to the piping and offer the advantage of a reduction in the number of joints. For pipe sizes under 5 in., autogenously welded nozzles are quite satisfactory, but for larger sizes the joints are usually **hammer-welded** (forged). Improvements in autogenous and electric welding are to be expected, and the above statements are necessarily limited to the present (1924) state of the art.

The Use of Welding in Power-plant Piping· Power, May 15, 1925, p. 680.

TABLE 94

THERMAL EXPANSION OF A FEW STEELS

(Bureau of Standards, Bul 433, 1922)

Composition	Steel Number 1	2	3	4	5
Si.	007	086	23	1 61	.25
Mn. .	06	64	68	57	.92
Sul. ...	.035	061	025	033	.033
Phos. .	012	.052	.012	.013	024
Carbon	.25	41	42	.44	.59
Temperature Range, Deg Fahr.	**Mean Coefficient of Expansion × 100,000**				
77°–212°	6 18	6 18	5 23	6 23	6 18
77°–572° .	6 95	7.07	6 74	7 07	7 18
77°–1112° .	7 95	8.05	7 95	8 05	8 11

302. Expansion of Pipes. — One of the most difficult problems in the design of a piping system is the proper provision for expansion and contraction due to change in temperature. If a pipe is under no stress when cold, and the temperature is increased, it will increase in length. The length will also be increased by the tensile stress effected by any internal pressure to which the pipe is subjected. The increase in length due to pressure is negligible except for extremely high pressures and long lengths of thin pipe, but that due to temperature may be considerable.

TABLE 95

EXPANSION OF PIPE

Increase in Length, Inches per 100 ft.

(Crane Co.)

Temp., Deg. Fahr.	Wrought Steel	Wrought Iron	Cast Iron	Brass and Copper
0	0	0	0	0
50	0.37	0.37	0.32	0.55
75	0.56	0.56	0.51	0.84
100	0.75	0.80	0.70	1.15
150	1.17	1.25	1.07	1.82
200	1.60	1.65	1.50	2.40
250	2.07	2.12	1.87	3.02
300	2.50	2.60	2.35	3.75
350	2.97	3.15	2.80	4.50
400	3.45	3.65	3.30	5.25
450	4.07	1.32	3.87	6.12
500	4.70	1.90	4.15	7.05
550	5.32	5.55	5.07	8.05
600	6.00	6.25	5.70	9.05
650	6.72	7.02	6.40	10.17
700	7.50	7.85	7.15	11.40
750	8.37	8.77	7.97	12.67
800	9.30	9.75	8.90	11.10

The increase in length for both conditions may be expressed

$$l_p = paL/EA \tag{268}$$
$$l_t = \mu(t_1 - t)L \tag{268a}$$

in which

l_p = increase in length due to pressure, in.,
l_t = increase in length due to the temperature difference,
p = pressure difference between inside and outside of pipe, lb. per sq. in. gage,
a = inside area of the pipe, sq. in.,
L = length of the pipe, in.,
E = modulus of elasticity (average for pipe steel = 30,000,000),
t_1 = final temp., deg. fahr. (the temperature of the pipe is practically that of the steam),
t = initial temp., deg. fahr.,
μ = mean coefficient of expansion between temperature t and t_1,
A = sectional area of the pipe material, sq. in.

Example 81. — A 12-in. extra heavy high-pressure steam main is 100 ft. long when cold (70 deg. fahr.): Required the increase in length when

carrying superheated steam at 250-lb. gage pressure, temperature 750 deg. fahr.

Solution. — Here $p = 250$, $a = 108.4$, $L = 1200$, $E = 30{,}000{,}000$, $A = 19.25$, $t_1 = 750$, $t = 70$, $\mu = 0.0000096$ (interpolated from data in Table 95).

Substituting these values in equations (268) and (268*a*),

$$l_p = \frac{250 \times 108.4 \times 1200}{30{,}000{,}000 \times 19.25} = 0.056 \text{ in., which is negligible,}$$

$$l_t = 0.0000096\ (750 - 70)\ 1200 = 7.8.$$

If the expansion of the pipe is constrained, as by anchoring both ends, an axial force will be exerted on the anchors which is equivalent to the tensile stress resulting from stretching the pipe at constant temperature the full amount of the expansion. This force is independent of the length of pipe and directly proportional to the increase of temperature. Unless well braced throughout its length, the pipe may buckle and become distorted. The axial force exerted by the temperature increase may be calculated from the following equation

$$P = EAl_t = EA(t_1 - t)\mu. \qquad (269)$$

Notations as in equations (268) and (268*a*).

Example 82. — A 6-in. "extra heavy" steel pipe is heated from 66 to 366 deg. fahr. (the temperature corresponding to steam at 165 lb. per sq. in. abs. pressure); required the axial force exerted if the pipe is constrained against movement in any direction.

Solution. — $E = 30{,}000{,}000$; $t_1 = 366$; $t = 66$; $\mu = 0.000007$ (interpolated from Table 95), $A = 8.5$ sq. in.

Substituting these values in equation (269), and solving

$$P = 30{,}000{,}000 \times 8.5\ (366 - 66)\ 0.000007 = 535{,}500 \text{ lb.}$$

Since the forces produced by expansion are practically irresistible, the pipe is invariably allowed to expand and its movement is prevented from unduly stressing the fittings and connections by

1. Long radius bends.
2. Double-swing screwed fittings.
3. Expansion joints.

Long-radius bends which utilize the elasticity of the pipe itself are commonly used for high-pressure work in preference to all other means for relieving the stresses due to expansion. The use of pipe bends reduces internal friction by providing easy turns, eliminates unnecessary fittings and joints, and facilitates clearing all other pipes and structural inter-

ferences. Pipe manufacturers are equipped to make special bends conforming to any shape or dimensions to which it is practical to bend pipe.

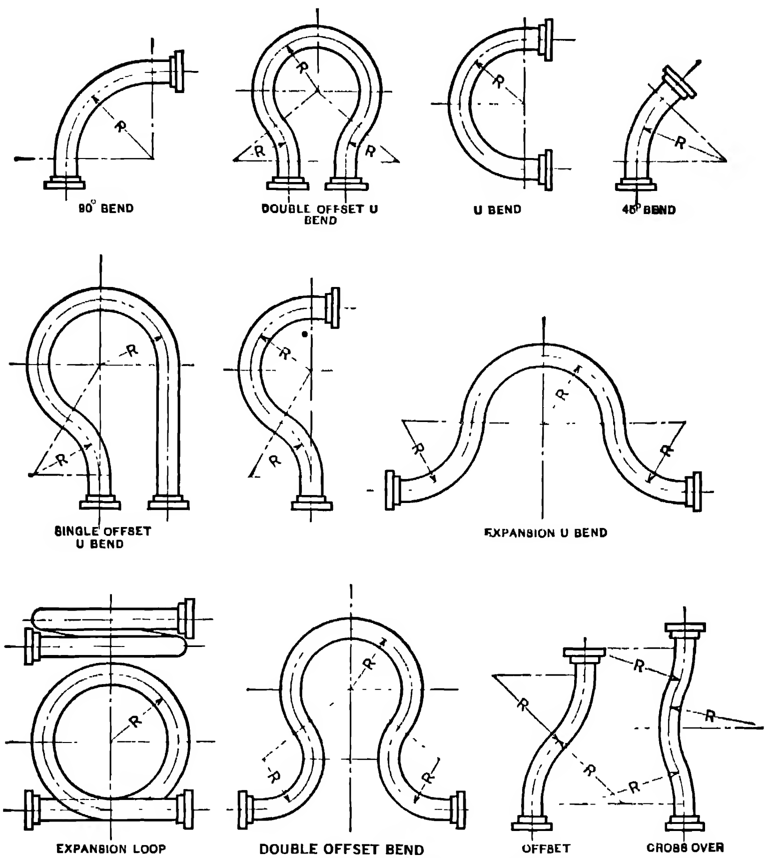


FIG. 528. Types of Standard Expansion Bends.

Figure 528 shows a number of standard bends. Table 97 gives the minimum radii and length of straight pipe at the end of each bend, and Table 96 the amount of expansion absorbed by a standard 90-deg. quarter bend and other shapes as recommended by the Crane Company. For an excellent treatise on pipe bends in which the problem is analyzed from both a theoretical and practical standpoint, consult "Elasticity of Pipe Bends" by S. Crocker and S. S. Sandford, *Trans. A.S.M.E.*, Vol. 44, 1922. This paper contains graphical charts by means of which the forces acting

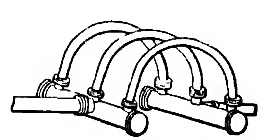


FIG. 529. U Bends for Large Headers with Limited Overhead Space.

against the anchorage, the maximum fiber stress occurring in the bend, and the amount of expansion which can be absorbed for a given stress may be obtained without calculation for standard pipe up to 20 in. and bent to radii up to 120 in.

TABLE 96

SAFE EXPANSION VALUES OF 90-DEGREE WROUGHT STEEL BENDS IN INCHES

(Full weight or extra heavy pipe)

(Crane Co.)

Sizes	Mean Radius of Bend (in Inches)												
	12	15	20	30	40	50	60	70	80	90	100	110	120
1	1/4	3/8	1/4	1 1/4	3 1/8								
2	1/8	1/4	1/2	1	1 3/4	2 1/4	3 7/8	5 3/8					
2½		1/4	3/8	7/8	1 1/2	2 1/4	3 1/4	4 1/2	5 1/4				
3		1/8	1/8	5/8	1 1/8	1 7/8	3 5/8	3 5/8	4 3/4	6			
3			1/4	5/8	1	1 5/8	2 1/8	3 1/8	4 1/8	5 1/4			
4			1/4	1/2	1	1 1/2	2	2 7/8	3 1/4	4 1/4	5 3/4		
4½				1/2	7/8	1 1/8	1 7/8	2 1/2	3 3/8	4 1/4	5 1/4		
5				3/8	3/4	1 1/8	1 5/8	2 1/4	3	3 1/4	4 5/8	5 5/8	
6				1/8	5/8	1	1 3/8	1 7/8	2 1/2	3 1/8	3 7/8	4 3/4	5 5/8
8					1/2	3/4	1	1 1/2	1 7/8	2 1/2	3	3 5/8	4 3/8
10						5/8	7/8	1 1/8	1 1/2	2	2 3/8	2 7/8	3 1/2
12							3/4	1	1 1/8	1 5/8	2	2 1/2	2 7/8
14								7/8	1 1/8	1 1/8	1 1/4	2 1/8	2 1/2
15									1	1 3/8	1 5/8	2	2 3/8
16									7/8	1 1/4	1 1/2	1 7/8	2 1/4
18												1 5/8	1 7/8
20													1 3/4

For U bend multiply expansion values by 2.
For Single Offset and Expansion U multiply by 4.
For Double Offset and Expansion Loop multiply by 5.

TABLE 97

MINIMUM DIMENSIONS FOR PIPE BENDS

(Crane Co.)

Size of Pipe, In.	Radius of Bend, In		Lengths of Straight Pipe on Each Bend, In	Size of Pipe, In.	Radius of Bend In		Lengths of Straight Pipe on Each Bend, In.
	Full Weight Pipe	Extra Heavy Pipe			Full Weight Pipe	Extra Heavy Pipe	
2½	12.5	7	4	8	40	28	9
3	15.0	8	4	9	45	35	11
3½	17.5	10	5	10	50	40	12
4	20.0	12	5	12	60	50	14
4½	22.5	14	6	14	70	65	16
5	25.0	15	6	15	75	70	16
6	30.0	20	7	16	80	78	18
7	35.0	24	8	18	108	88	18

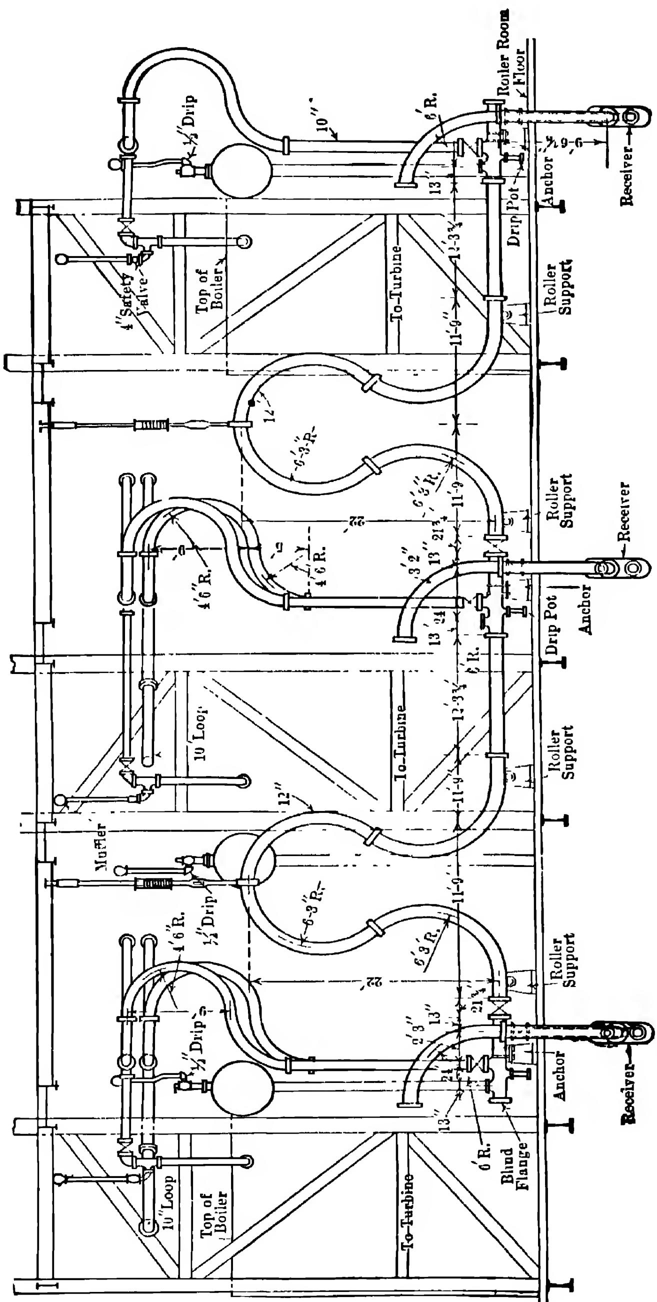


FIG. 530. Typical Expansion Bends. Buffalo General Electric Co.

Considering a pipe bend as a beam of special form one end of which is fixed and the other end of which is free to move in the direction of the acting force, it can be shown[1] that

$$d = KFR^3/EI \tag{270}$$

$$S = CDdE/R^2 \tag{270a}$$

in which

d = deflection of the free flange measured in the direction of the force F, in.
F = force acting against the flange, lb.
R = radius of the bend, in.
E = modulus of elasticity, lb. per sq. in.
I = moment of inertia of the pipe section, in.[4]
S = maximum fiber stress in the bend, lb. per sq. in.
D = outside diameter of the pipe, in.
K, C = constants. See Table 98.

Example 83. — What must be the radius of a standard expansion **U** bend made up of 10-in. extra heavy pipe in order to absorb an expansion of 1.5 in. and exert a pressure of 600 lb. on the anchorage. Required also the maximum fiber stress. Assume E = 30,000,000.

Solution. — From pipe tables, we find the outside and inside diameter of an extra heavy steel pipe to be 10.75 and 9.75 in., respectively. The moment of inertia of the section is $I = 0.049\ (10.75^4 - 9.75)^4 = 216$.

TABLE 98

VALUES OF CONSTANTS K AND C

Type of Bend	Direction of Force, Fig. 531	K	C	Type of Bend	Direction of Force, Fig. 531	K	C
Quarter	a	0.36	1.40	Expansion U	d	9.4	0.106
Quarter	b	0.88	0.64	Double offset	e	40.0	
Plain U	c	1.57	0.32				

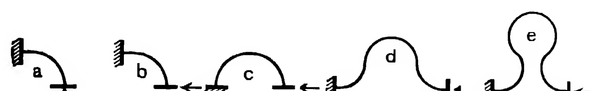


FIG. 531. Direction of Force (Table 98).

Substituting I = 216; E = 30,000,000; F = 600; d = 1.5 and K = 9.4 (from Table 98) in equation (270) and solving, we have

$$1.5 = 9.4 \frac{600R^3}{30{,}000{,}000 \times 216}$$

$$R = 120 \text{ in.}$$

[1] Trans. A.S.M.E., Vol. 44, 1922, p. 547.

From equation (270a)

$$S = 0.106 \frac{10.75 \times 1.5 \times 30{,}000{,}000}{120^2}$$
$$= 3560 \text{ lb. per sq. in.}$$

Figure 532 shows a **double-swing screwed joint** in which expansion causes the fittings to turn slightly and thus relieve the strain. This method is usually adopted in low-pressure heating systems where the pipes are small and only a small amount of expansion takes place. It is wholly unsuited for large pipes and high pressures. **Packed expansion** or **slip joints,** Fig. 533, are occasionally used for high-pressure steam service where space requirements prohibit the installation of long-radius bends, but they are objectionable because of the possibility of leakage and sticking. In very long lines of straight piping, slip joints are frequently the only solution of the expansion problem. When slip joints are employed, the pipe must be securely anchored to prevent the steam pressure from forcing the joint apart and at the same time permit the pipe in expanding to work freely in the joint. Sagging of the pipe on either side, which may cause binding in the joint, is prevented by suitable supports. Balanced-pressure slip joints have been developed but they are not much in evidence. For pressures below 150 lb. gage, corrugated or otherwise flexibly fabricated copper expansion joints are in common use, and in condenser service the condenser is frequently connected to the turbine exhaust opening by means of rubber expansion joints. For a detailed description of such an expansion joint see Report of Prime Movers Committee, T5-21, N.E.L.A., 1921, p. 10.

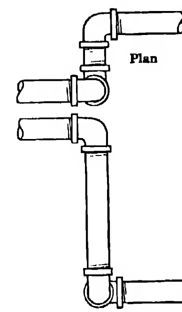


Fig. 532. Double-swing Expansion Joint.

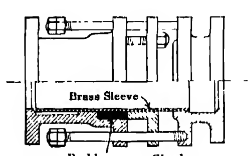


Fig. 533. Slip Expansion Joint.

303. Pipe Supports and Anchors. — Pipe lines must be supported to guard against excessive stresses caused by dead weight, expansion, and vibration. There is no standard practice in this connection and the means adopted by different plants vary over a very wide range. In order to limit the thrust exerted by the piping on the throttle of prime movers or other stationary points of attachment, fixed anchorages in the steam system, adjacent to the joints where the leads come off the header, are usually necessary. In the majority of installations the anchors are rigidly attached to the building structure and the balance of the piping

is supported at various points by **hangers, wall brackets,** or **floor stands.** Figure 534 illustrates a common design of anchor of the wall-bracket type, suitable for moderate end thrusts. The pipe rests upon a saddle

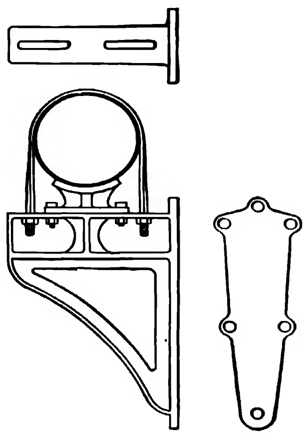


FIG. 534. Typical Pipe Anchor.

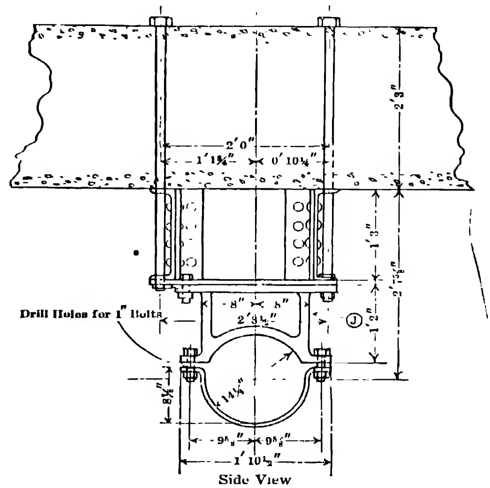


FIG. 535. Pipe Anchor — Union Electric Light & Power Co.

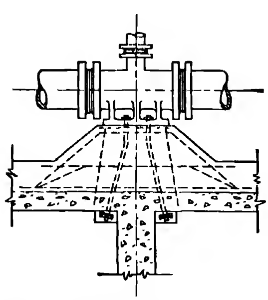


FIG. 536. Pipe Anchor — Lakeside Station.

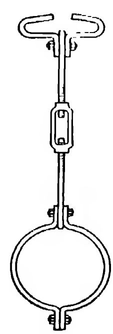


FIG. 537. Typical I-beam Hanger.

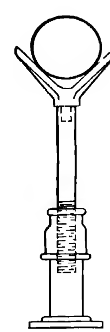


FIG. 538. Typical Adjustable Floor Stand.

and is rigidly clamped to the bracket by a flat iron band with ends threaded and bolted. Figure 535 shows the manner in which the 12-in. steam main is anchored to the building structure in the power house of the

Union Electric Light and Power Co. and Fig. 536 shows the method of anchoring the main header in the Lakeside Power Plant of the Milwaukee Elec. Ry. and Light Co. Anchor attachments are also welded to the pipe.

Figure 537 illustrates a convenient type of flexible hanger for suspending pipes from " I " beams. This design is suitable only for supporting the dead weight of the pipe and where free movement is permissible. Where the movement is to be constrained in an axial line, the pipe is held between two rollers as shown in Fig. 539 or supported on a sliding guide

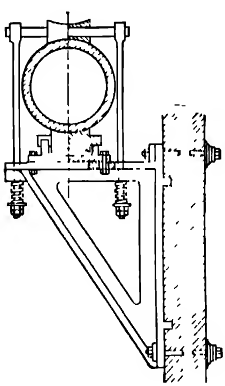


FIG. 539. Typical Wall Bracket with Binding Roll.

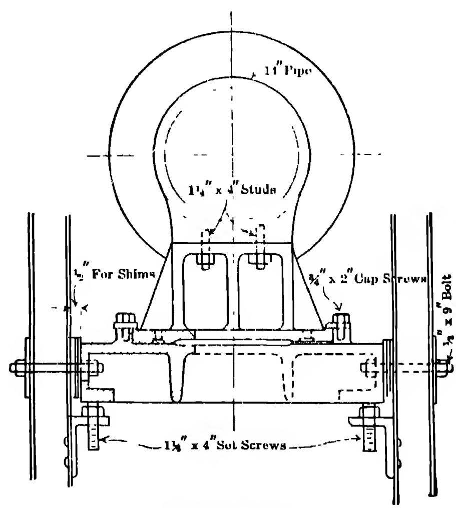


FIG. 540. Pipe Support and Guide — Marysville Power House.

as in Fig. 540. For straight runs of pipe the support is usually of roller construction but, where there are branch connections or bends leading from a main, some design of axial guide support is necessary to prevent the pipe from springing laterally. Figure 541 illustrates a method of suspending and counter-balancing expansion loops in a main header and Fig. 542 a flexible support for a large vertical exhaust header. Other arrangements of piping, supports, and anchors will be found in this chapter. Pipe supports and anchors should be designed so that they can be readily removed without disturbing the pipe line and should be adjustable to facilitate " lining up."

Steam Pipe Supports: Power Plant Engrg., Dec. 1., 1919, p. 1055.

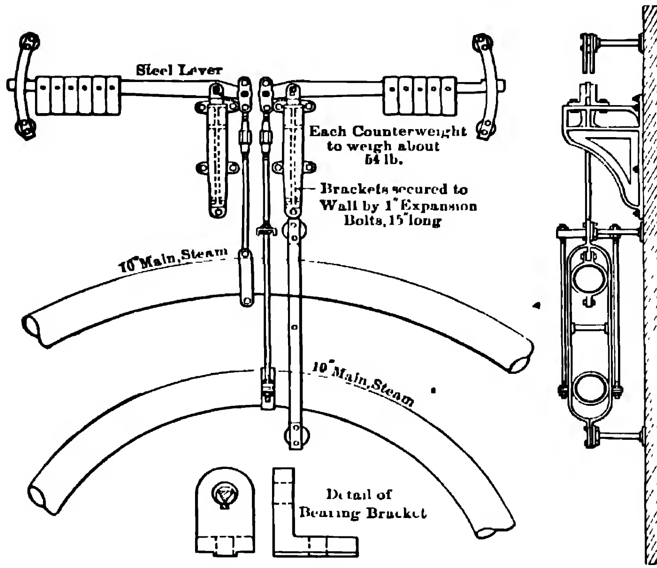


FIG. 541. Method of Suspending and Counter-balancing Expansion Loops in Steam Mains.

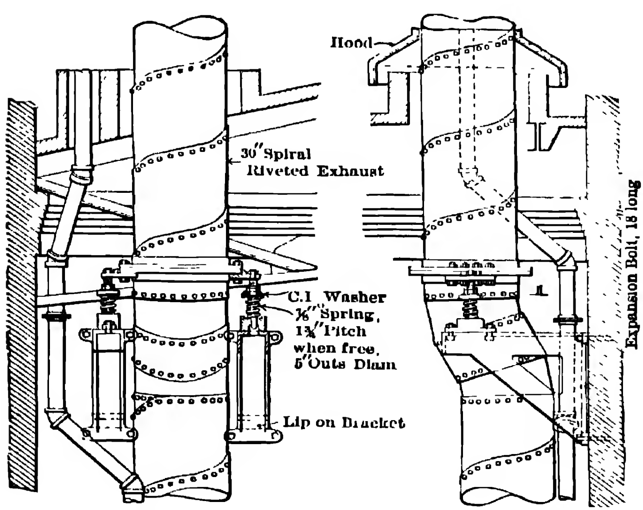


FIG. 542. Spring Support for 30-inch Exhaust Pipe.

304. Loss of Heat from Bare and Covered Pipe. — Steam pipes, feed-water pipes, boiler steam drums, receivers, separators and the like should be covered with heat-insulating material to reduce heat losses to a minimum. By properly applying any good commercial covering, from 75 per cent to 95 per cent of the heat loss may be prevented. Numerous investigations have been made relative to the heat losses from bare and covered pipes, but the results have been far from harmonious. The most trustworthy results are those based upon the investigations of J. C. Peebles (Research Laboratory, Armour Institute of Technology), L. B. McMillan (*Trans. A.S.M.E.*, Vol. 37, 1915, p. 921), Bagley (*Trans. A.S.M.E.*, Vol. 40, 1918, p. 667), and R. H. Heilman (*Trans. A.S.M.E.*, Vol. 44, 1922, p. 299). The results of these investigators agree as closely as can be expected, considering the variation in the structure of the different samples from the same commercial material and the influence of the surroundings in the laboratories in which the tests were conducted. From the investigations of Heilman on bare pipe, Fig. 543, it will be seen that the heat loss from bare pipes conducting heated fluids is so great that any good grade of pipe covering may pay for itself in a comparatively short time. The curves in Fig. 544, based upon the results of McMillan's investigation, which check substantially with the tests of Peebles and Heilman, give the heat loss per deg. fahr. temperature difference per sq. ft. per hr. for a number of commercial classes of coverings, for various temperature differences between the temperature of the pipe surface and that of the surrounding air. While the maximum temperature difference is limited to 500 deg. fahr., which is considerably lower than that incurred in the modern central station, the heat loss at the higher figures may be obtained with sufficient accuracy for most purposes by extending the curves.

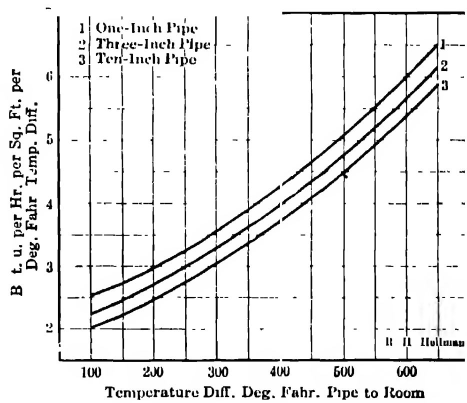


FIG. 543. Heat Loss from Bare Pipe.

If

H_2 = heat loss in still air per sq. ft. of outside covering surface, B.t.u. per hr.,

k = conductivity of the material, B.t.u. per hr. per sq. ft. per in. thickness per deg. temperature difference between the outer and the inner surface of the covering,

t_2, t_1 and t = temperatures, respectively, of the outer surface of the covering, pipe, and air in the room, deg. fahr.,

r_2 and r_1 = radii, respectively, of the outer and the inner surface of the covering, in.,

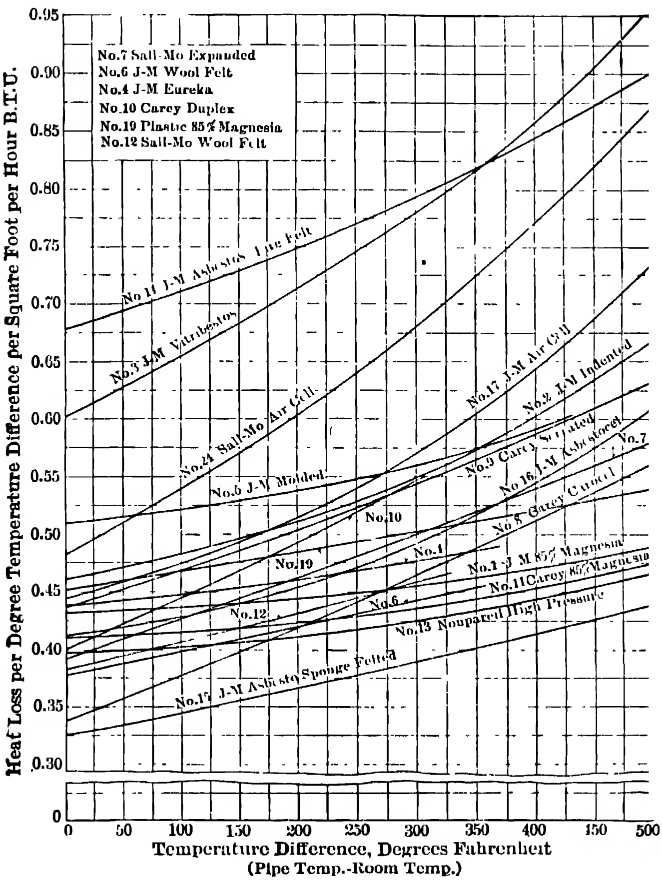


FIG. 544. Heat Loss through Pipe Coverings (Single Thickness).

d = temperature difference between the covering and air corresponding to a rate of loss H_2,

H_1 = heat loss per sq. ft. of pipe surface, B.t.u. per hr.,

x = thickness of covering for flat surfaces with parallel planes.

it can be shown (*Trans. A.S.M.E.*, Vol. 37, 1915, p. 960) that for

Flat Surfaces

$$H_2 = k\,(t_2 - t_1)/x \tag{271}$$

Cylindrical Surfaces

$$H_2 = k\,(t_1 - t - d)/r_2\,(\log_e r_2 - \log_e r_1) \qquad (271a)$$
$$= r_1\,H_1/r_2 \qquad (271b)$$
$$k = H_1 r_1\,(\log_e r_2 - \log_e r_1)/(t_1 - t_2). \qquad (271c)$$

The curve in Fig. 545 gives the value of k for 85 per cent canvas-covered magnesia as determined by Bagley. Curves of this nature for various insulating materials greatly simplify the calculation of the heat losses. The curves in Fig. 546, showing the relation between H_2 and d as determined by Peebles, McMillan, and Heilman, offer a means of calculating the value of k from test data as shown in Fig. 544 but unfortunately these curves show considerable departure from each other for the same temperature conditions. Curves (1) and (2), Fig. 546, give satisfactory results for pipe lines protected against air currents but for exposed lines preference should be given to curves (3) and (4).

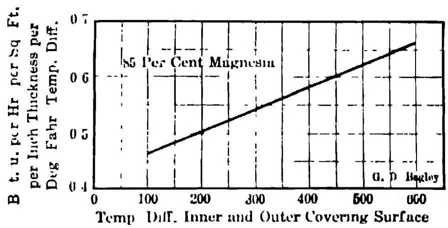


Fig. 545. Coefficient of Thermal Conductivity.

Applications of equations (271) to (272c) are best illustrated by examples 84 and 85.

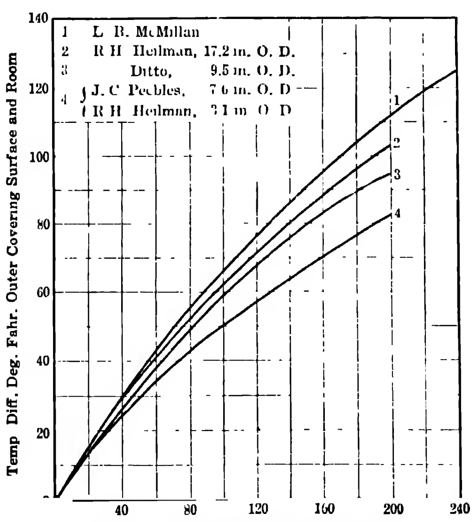


Fig. 546. Relation between Heat Loss and Temperature Difference.

Example 84. — A steam pipe, 5.6. in. outside diameter, is covered with single-thickness J-M 85 per cent magnesia, 1.13 in. thick, temperature of the pipe 380 deg. fahr., room temperature 80 deg. fahr. Required the conductivity per in. thickness for the given conditions.

Solution. — From Fig. 544 the rate of heat loss per hr. per sq. ft. per deg. temperature difference is 0.455 B.t.u. Therefore, $H_1 = 300 \times 0.455 = 136.5$ and $H_2 = 136.5 \times 5.6/2 \div (5.6/2 + 1.13) = 97.2$ B.t.u. From Fig. 546 (A) the temperature difference between outer covering surface and air corresponding to a loss of 97.2 B.t.u. is 65 deg. fahr. Therefore, the temperature difference between the inner and the outer covering surface is 300 −

65 = 235 deg. fahr. Substituting these values in equation (272*c*) and solving for k

$$k = [136.5 \times 2.8\ (log_e\, 3.93 - log_e\, 2.8)] \div 235 = 0.551.$$

Example 85. — If the pipe in Example 84 is covered with 3-in. thickness of material, other conditions remaining the same, calculate the heat loss per sq. ft. of pipe surface per hr. per deg. temperature difference.

Solution. — From equation (271*a*)

$$H_2 = \frac{0.551\ (380 - 80 - d)}{(2.8 + 3)\ (log_e\, 5.8 - log_e\, 2.8)}$$
$$= 0.13\ (300 - d).$$

Now assume d = 20 deg. Then from Fig. 546 — Curve No. 1 H_2 = 25.5 B.t.u. But from equation (271*a*) H_2 = 0.13 (300 − 20) = 36.4. This shows that d must be greater than 20. Assume d = 30. Then from Fig. 546 H_2 — Curve No. 1 = 39.5 B.t.u. and from equation (271*a*) H_2 = 0.13 (300 − 30) = 35.1. This shows that d must be less than 30. By cut and trial the correct value d_2 = 27 may be obtained. Then H_2 = 0.13 × (300 − 27) = 35.5. Substitute this value of H_2 in equation (271*b*) and solve for H_1

$$35.5 = 2.8 H_1 \div 5.8$$

from which H_1 = 73.5 B.t.u. per hr. per sq. ft. Loss per sq. ft. per hr. per deg. temperature difference between the pipe surface and air in the room = 73.5 ÷ 300 = 0.245 B.t.u.

The pipe-covering materials most commonly found in the modern central station are **85 per cent magnesia, sponge felt, silocel** and **nonpareil.** Of these, 85 per cent magnesia is still the predominant material for temperatures up to 600 deg. fahr. For metal temperatures above 600 deg., it is desirable to use some form of heat-resisting insulation as a first layer in order to reduce the temperature of the inner surface of the standard insulations. Asbestos fiber matted and bound with silicate of soda has the necessary heat-resisting feature and has been extensively used, but its heat conductivity is relatively high. In some of the latest stations employing highly superheated steam, the metal surface is first covered with a 1-in. layer of **Carey " Hi-Temp "** or the equivalent which has the properties of high insulating efficiency and resistance to mechanical deterioration at temperatures up to 1000 deg. fahr., and this is then followed with a second covering 2 in. thick of 85 per cent magnesia or the equivalent.

Pipe covering is applied in sections molded to the required form and held to the pipe by bands, or may be applied in a plastic form. The former is more readily applied and removed, and is usually adopted for pipes, while the valves and fittings are generally covered with plastic

material. Piping should be tested under pressure before being covered, since leaks destroy the efficiency and life of the covering. If the surrounding atmosphere is moist the covering should be given two or three coats of good paint. Coverings are sometimes applied to cold-water pipe to prevent sweating.

Commercial Efficiency of Single and Graded Steam Pipe Covering: State College of Washington, Bul. No. 12, 1923.
Recent Developments in Pipe Insulation. Power Plant Engrg., July 1, 1924, p. 698.
Rational Design of Coverings for Pipes: Mech. Engrg., Oct., 1925, p. 805.

305. High-pressure Steam Piping Systems. — In the older stations, in which the prime movers were of the reciprocating type and the boilers of comparatively small capacity, the boiler and engine room were arranged back to back, end to end, or double-decked, according to the space available, and the high-pressure steam lines were arranged on the spider, single header, duplicate header, and loop or ring header systems. All of these arrangements and systems were more or less standardized and differed only in minor details. In the modern central station there is no standard arrangement of turbines and boilers, and the piping system is designed to meet each specific set of conditions.

Figure 547 shows the **back to back** arrangement of engines and boilers in which the engines and boilers are housed in separate rooms and the steam from each boiler is led to a common or main header. This was standard practice in the central station in the year 1905 and is still used in a number of small plants. The main header was placed in the boiler room along the division wall and it was extended as the growth of the station demanded. This system permits of short and direct connections between prime movers and boilers and is simple and compact. To insure continuity of operation in case of injury to the main header, **duplicate** or auxiliary headers were occasionally installed. Figure 548 shows a back-to-back arrangement in which the length of the main header is greatly shortened and the various distributing pipes lead directly to steam-using appliances. This is known as the **spider** system of piping. This system has given satisfactory results in small plants but is rather unsightly. The **loop header,** Fig. 550, has the advantage over the single header in that the steam supply may be taken from either end of the boiler battery, should occasion arise, or from both ends to insure uniform boiler operation. The extra length of main header increases the first cost and offers a larger surface for heat losses.

Figure 551 illustrates a typical installation in which the boiler and engine room are **end to end.** This is a common arrangement where only a narrow strip of space is available for the plant.

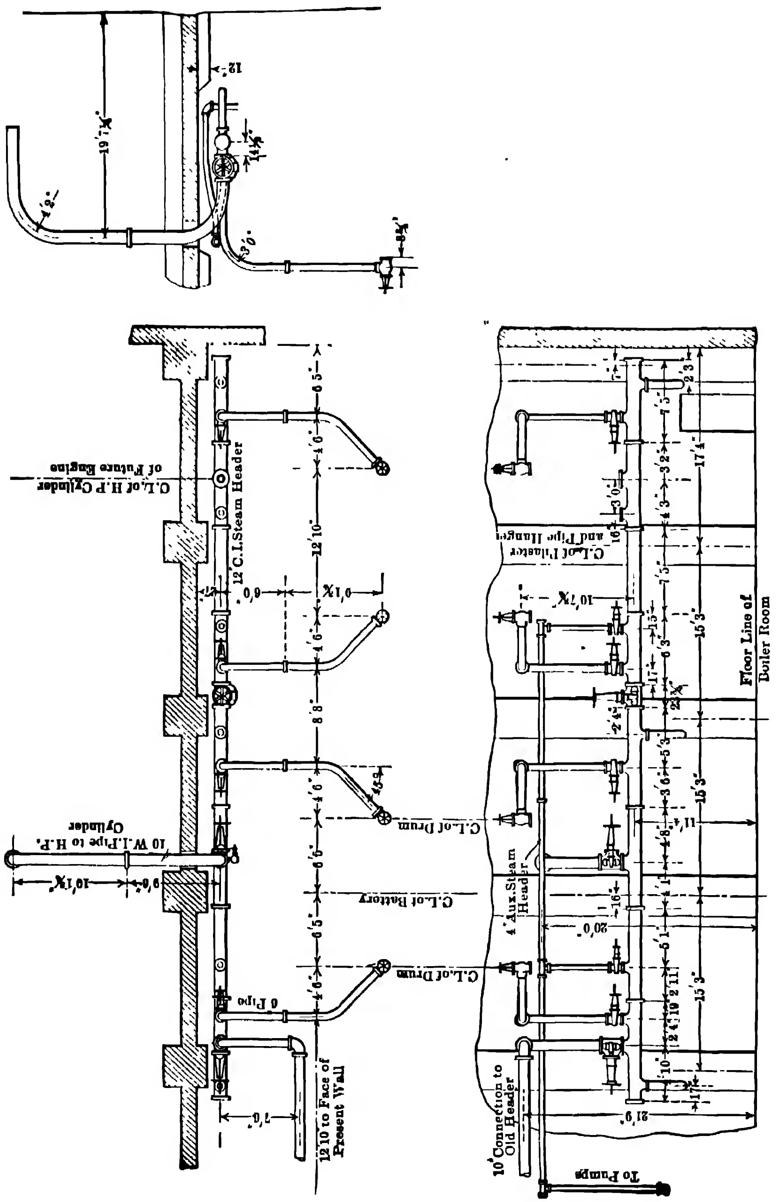


FIG. 547. Typical "Back to Back" Arrangement with Single Main Header.

Double- and even **triple-deck** installations, in which the boilers and prime movers are on separate floors, are to be found in a few cases where ground space is very costly, but the first cost of such a plant is very high.

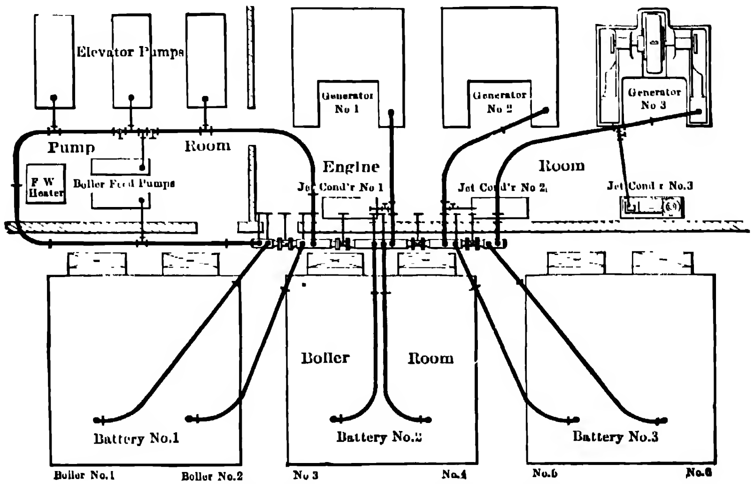


FIG. 548. Typical "Spider" System.

The Highland Park plant of the Ford Motor Co. is an excellent example of the double-decked arrangement.

The modern central station is usually designed on the **unit** basis in which each turbo-generator has its own boiler and auxiliary equipment which

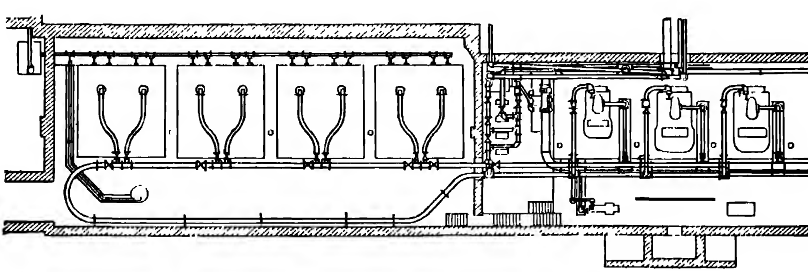


FIG. 549. Typical Auxiliary Header System.

may be operated independently of the rest of the plant. The steam mains and many of the auxiliaries are cross-connected so as to supply energy to other units in case of emergency, but to all intents and purposes each unit is an independent plant.

Figure 552 shows the arrangement of boilers and turbines in the Yonkers Power House of the New York Central, illustrating standard practice of a

decade ago. The turbines are arranged side by side in a single room separated from the boiler housing by a division wall. The turbines are

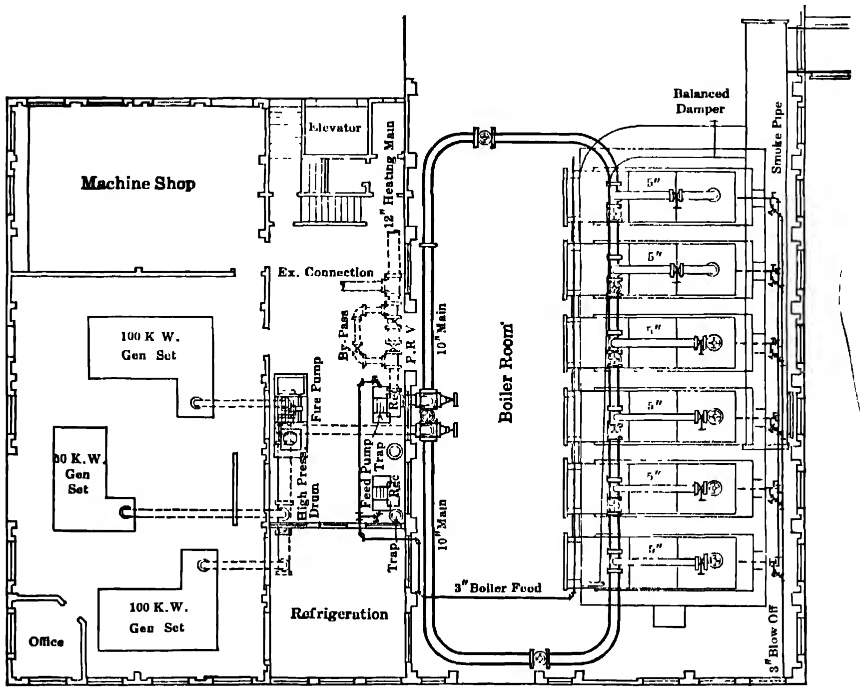


FIG. 550. Typical "Loop Header" System.

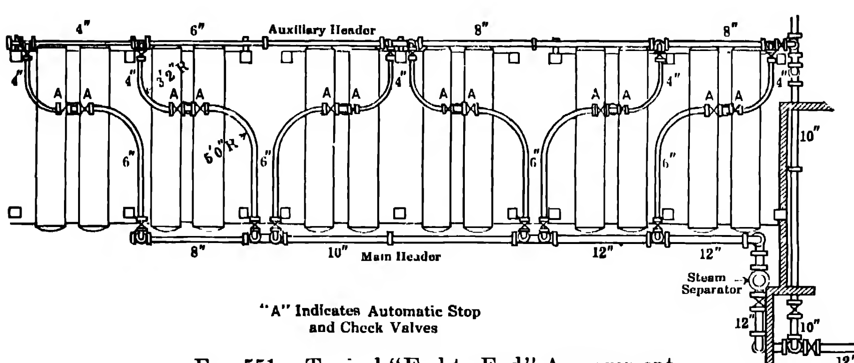


FIG. 551. Typical "End to End" Arrangement.

connected in pairs by 14-in. loops, each turbine taking steam from either of two banks of four boilers. The high-pressure piping for each pair of boilers is cross-connected to the adjacent pair by a cross-over main.

Figure 553 shows the arrangement of boilers, turbines, and high-pressure steam piping in the first section of the new Hell Gate Station. The tur-

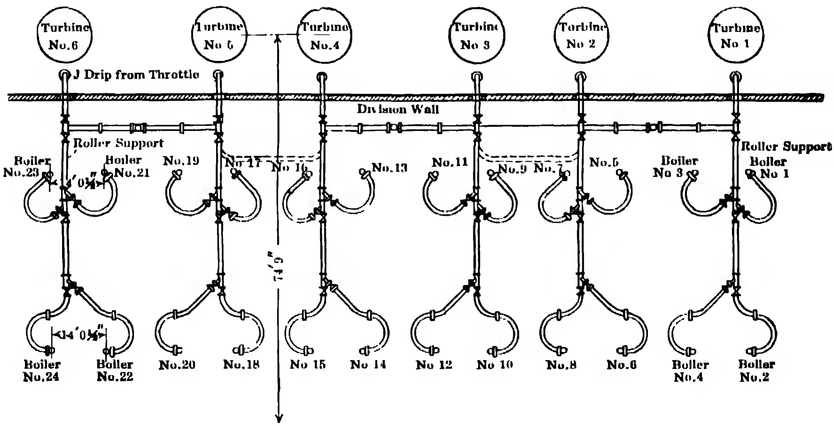


FIG. 552. Typical "Unit" System.

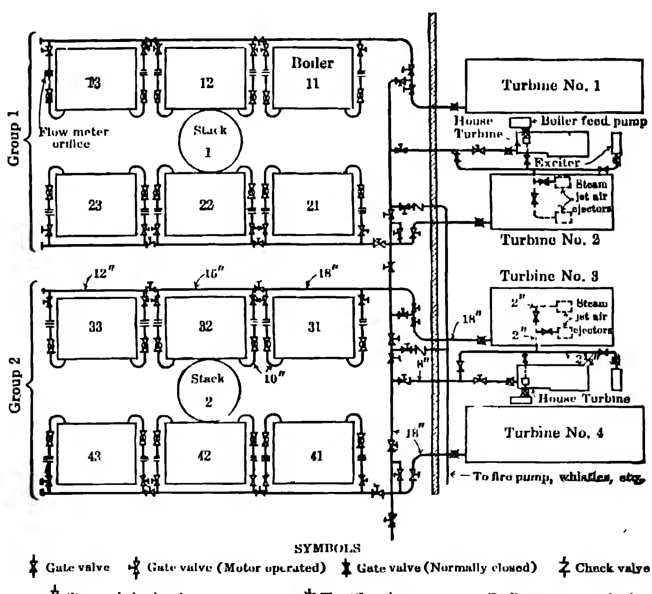


FIG. 553. High-pressure Piping — Hell Gate Station.

bine units and boilers are grouped in pairs and each pair is cross-connected as indicated. Each group of six boilers has a single stack and breeching. In order to insure continuity of operation, such elements of the various

groups as forced-draft air supply, feedwater supply, etc. may be interconnected at will.

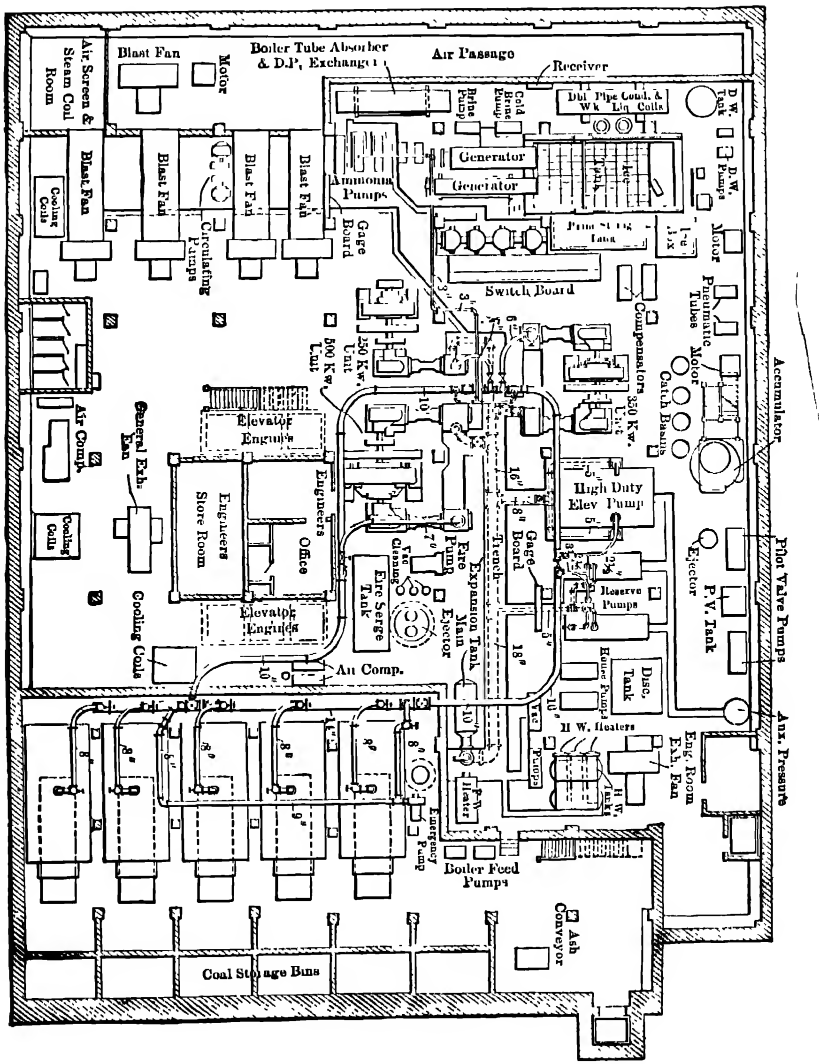


FIG. 554. General Arrangement of Steam and Exhaust Piping. La Salle Hotel, Chicago.

Figure 554 shows the general arrangement of the piping in the La Salle Hotel, Chicago, illustrating the loop header as applied to this class of plant.

306. Exhaust-Steam Piping. — In the large central steam-power station, there is no condenser exhaust piping, at least in so far as the prime mover is concerned, since the turbine exhaust flange is bolted directly to the condenser or indirectly through the agency of a short expansion joint. This applies also to the smaller plants equipped with individual condensers. Separate free-atmospheric exhaust lines lead from each condenser to the roof. Occasionally a number of condensing units discharge into a single condenser, in which case the main vacuum line is increased in diameter as it approaches the condenser, the increase corresponding to the added weight of exhaust steam. In some designs, there is a main atmospheric exhaust line connected to the individual atmospheric exhaust branches, while in others a single atmospheric relief valve at the condenser suffices for all units.

In the majority of non-condensing plants, all or a part of the exhaust steam is used for heating or other industrial purposes, in which case an elaborate system of exhaust piping may be necessary. The general arrangement of apparatus in a typical non-condensing plant utilizing the exhaust for heating purposes is shown diagrammatically in Fig. 555 and the principles of operation are described in paragraph 3. The chief requirements for a combined power and exhaust-steam heating system are: (1) minimum back pressure on the prime mover; (2) effective and continuous drainage of condensation from supply pipes and radiators; (3) continuous removal of air and entrained moisture from confined spaces; (4) independent regulation of temperature in each radiator; (5) continuous return of condensation to the boilers; (6) utilization of part of the exhaust for preheating the feedwater; and (7) automatic regulation. The principal factor in any system of exhaust-steam heating is the trap or automatic outlet valve attached to each radiator or heating coil which permits both the water of condensation and the non-condensable gases to be removed automatically without building up back pressure. The heat given off by the radiators may be regulated by varying the quantity of steam supplied, either by hand or automatically by thermostatic control. Figure 511 shows the type of trap commonly employed in current practice and Fig. 577 shows a section through a popular design of thermostat for automatically opening and closing the exhaust steam admission valves to the radiators. (See paragraph 296.)

The main exhaust header in Fig. 555 is in the basement and the branch supply pipes feed upward. In tall office buildings the exhaust main frequently leads to the attic where it is connected with a distributing header and the branch supply pipes feed downward. While the latter arrangement is the better from a circulating standpoint, it requires additional

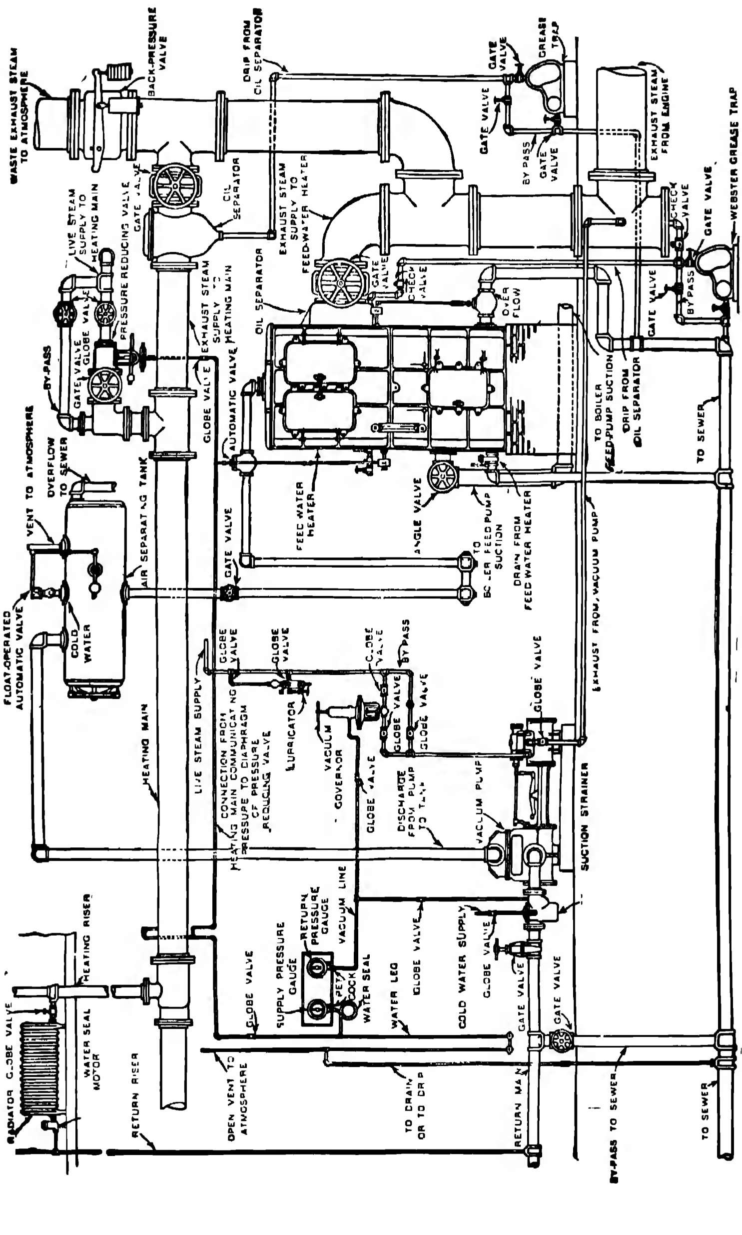


FIG. 555. Exhaust Piping. Combined Steam Power and "Vacuum" Heating Plant.

space for its installation and the back-pressure valve is remote from the engine room.

In industrial plants where the power requirements are large and the exhaust from the prime movers is too great to permit of discharge to waste, the engines or turbines are operated condensing (provided the cost of circulating water is not prohibitive) and steam for heating is extracted at a suitable point between the high-pressure and low-pressure stages. The exhaust, after it leaves the prime mover, is handled in the same manner as in the non-condensing plant.

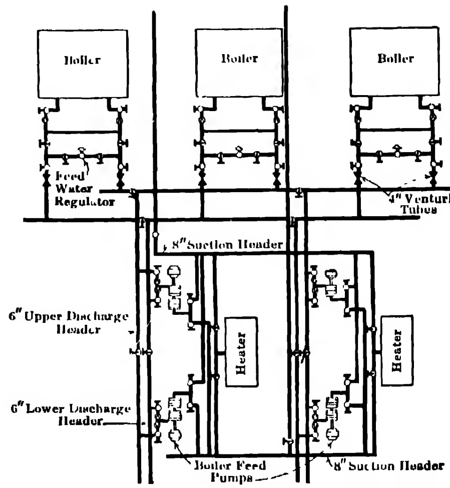


FIG. 556. Feedwater Piping. Dodge Bros. Power House.

307. Feedwater Piping. — The simplest arrangement of feedwater piping may be found in the small non-condensing plants, in which the feedwater is obtained under a slight head, such as is afforded by the average city supply, and is heated in an open heater by the exhaust steam from the engine to a temperature varying from 180 to 220 deg. fahr. depending upon the back pressure maintained on the heater. The hot feedwater gravitates from the heater to the pump and then is forced to the

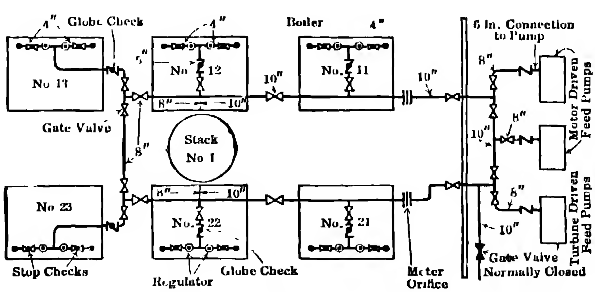


FIG. 557. Feedwater Piping at Hell Gate. Group 1.

boiler or to the economizer if one is used. If a meter is used, it is generally placed on the discharge side of the pump, and should be by-passed to permit it to be cut out for repairs. Plants operating continuously should have feed pumps in duplicate. In some cases, the returns from the heat-

ing system gravitate to the heater and only enough cold water is added to make up the loss from leakage, etc. In other cases, the returns gravitate to a special "returns tank," from which they are returned directly by pump or trap to the boiler without further heating. Occasionally a live-steam purifier is used, especially if the water contains a large percentage of calcium sulphate. The feed is then subjected to boiler pressure and temperature and the greater part of the impurity precipitated before it enters the boiler. Closed heaters are often used instead of open heaters. When the supply is not under head, a closed heater is usually preferred and is placed between the pump discharge and the feed main.

In small condensing plants with steam-driven auxiliaries, the feed piping is similar to that in non-condensing plants, except that if exhaust steam is used for heating purposes, it is supplied by the auxiliaries, such as feed pumps, stoker drives, condenser engines, and other steam-using appliances.

In plants having a number of boilers, it is customary to run a feed main or header the full length of the boiler room and connect it to each boiler by a branch pipe. This main may be a simple header, or it may be in duplicate, or of the "loop" or "ring" type. The feed main may run along the fronts of the boilers just above the fire doors, or above or under the settings, depending upon the design of the boiler room. Where a single header is used, the feed pumps are sometimes placed so as to feed into opposite ends of the main, which is then cut into sections by valves. Another arrangement is to place the pumps so as to feed into the middle of the header. With the loop arrangement the main is ordinarily cut into sections by valves, so that the water may be sent either way from the pumps and any defective section cut out. With duplicate mains a common arrangement is to place one main along the front of the boiler and the other at the rear or both overhead. Sometimes one main is placed in the passageway below the boiler setting and the other on top.

In the large central station, with its intricate system of obtaining a proper heat balance, the arrangement of the various heaters and feedwater piping system is one of the most difficult problems in the station design. Figure 556 shows the arrangement of the feedwater piping in the new Dodge Brothers power station, illustrating a comparatively simple layout. A diagrammatic outline of the feedwater piping at the Hell Gate Station is shown in Fig. 557, and Fig. 558 gives a similar view of the piping in the Hudson Ave. Station of the Brooklyn Edison Co. See also Figs. 415 to 418.

In the majority of modern plants the feedwater pipe lines are of wrought steel, but in some of the older plants, particularly where the water is of poor quality, the leads from header to boiler are of brass.

Figure 559, *A* to *E*, illustrates the various combinations of check valve, stop valves, and regulating valve in steam boiler practice. The simplest arrangement and one sometimes used in plants operating intermittently

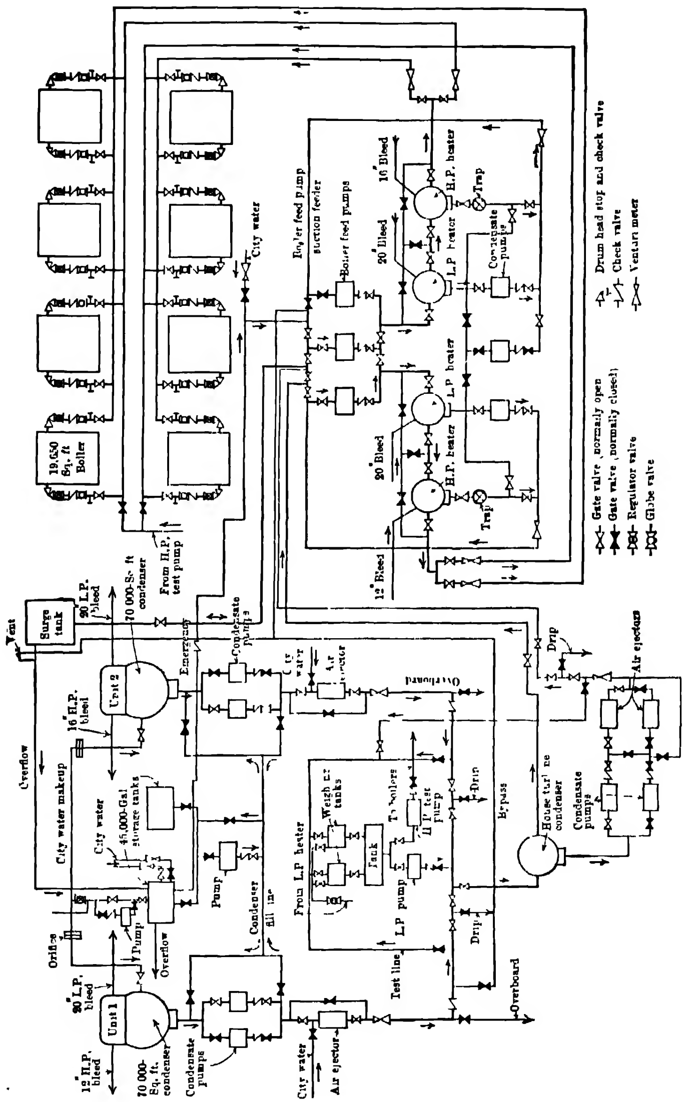


FIG. 558. Feedwater Piping. Hudson Avenue Station.

is shown in *A*. Here there are but two valves between the boiler and the main, the check being nearest the boiler and the stop valve at the main. The stop valve performs both the function of cutting out the boiler and

that of regulating the water supply. This arrangement is not recommended, as any sticking or excessive leaking of the check valve will necessitate shutting down the boiler. *B* shows the most common arrangement. Here the check valve is placed between the regulating valve and a stop valve as indicated. This permits a disabled check to be easily removed while pressure is on the boiler and the main. *E* shows an arrangement whereby both check and regulating valve may be removed, and is particularly adapted to boilers operating continuously where the regulating

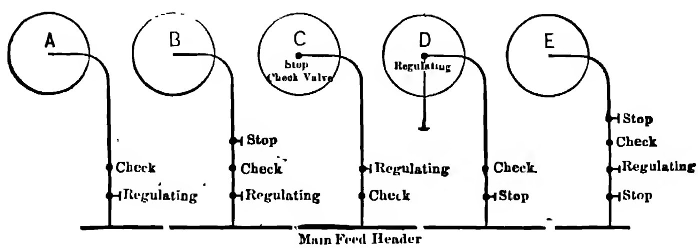


FIG. 559. Different Arrangements of Valves in Feedwater Branch Pipes.

valve is subjected to severe usage. In this case the stop valves are run wide open and are subjected to no wear. The regulating valve most highly recommended is a **self-packing** brass globe valve with **regrinding disc.** The check valve is ordinarily of the **swing check** pattern with regrinding disc, Fig. 581 (*C*).

Underground Steam Mains. Power, Apr 2, 1918, p. 460; Apr. 16, 1918, p. 540.

308. Flow of Steam in Piping System. — Notwithstanding the numerous investigations conducted on laboratory apparatus and on pipe lines under actual service conditions, all rules relative to the flow of steam in commercial piping systems are more or less empirical and are limited to the conditions under which the tests were conducted. Many of these rules give fairly satisfactory results when applied to straight pipes under 6 in. in diameter, free from obstructions, and for moderate pressures and temperatures. However, when applied to the large pipes employed in the modern central station with its high pressure and highly superheated steam, the results are apt to be seriously in error. While the existing rules for the flow of steam in straight pipes are more or less unsatisfactory, those pertaining to the flow in superheaters, valves, and fittings are even more so, and, considering the fact that the influence of the latter on the flow is usually much greater than that of the pipe itself, the designing engineer is forced to rely upon judgment and experience rather than upon theory. Practically all rules for the flow of steam in pipes are based upon the fundamental equation for the flow of compressible fluids (see Principles

of Thermodynamics, Goodenough, p. 161), and may be expressed as

$$p = Cv^2yL/d \tag{272}$$

$$p = Kw^2L/yd^5 \tag{272a}$$

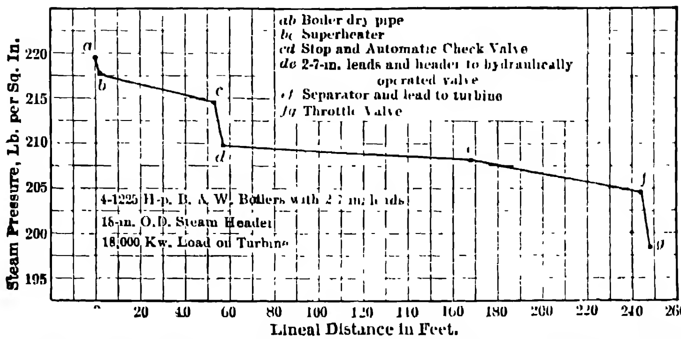


FIG. 560. Steam Pressure Drop from Boiler Drums to Turbine Throttle.

in which

p = pressure drop, lb. per sq. in.,
C, K = coefficients involving a number of reduction constants and including the coefficient of frictional resistance,
v = velocity of flow, ft. per sec.,
w = weight of flow, lb. per sec.,
y = mean density, lb. per cu. ft.,
d = internal pipe diameter, in.,
L = length of straight pipe or its equivalent, ft.

The values of C and K given by various investigators are given in Table 99.

TABLE 99

VALUES FOR COEFFICIENTS C AND K

Authority	Reference	100,000 C	K
Babcock	"Steam," 1919, p. 317	$1.13(1+3.6/d)$	$0.17(1+3.6/d)$
Carpenter...	Trans. A.S.M.E., Vol. 20, p. 347	$1.1(1+3.6/d)$	$0.475(1+3.6/d)$
Eberle...	"Eng. Thermo.," Lucke, p. 115	2.67	0.90
Fritzsche...	Mitt. uber Forschungsarbeit, Vol. 60	$5.16/d^{0.269} \times v^{0.118}$	$0.8d^{0.03}/w^{0.15}$
Gutermuth	Zeit. d. Ver. D. Ingr., Apr. 16, 1904, p. 572	3.82	1.28
Martin...	Engrg. (Lond.) Mar. 19, 1897	$1.43(1+3.6/d)$	$0.47(1+3.6/d)$
Hawksley .	Pro. Inst. Civ. Eng., Vol. 33, p. 55	3.60	1.21
Spitzglass...	Armour Engr., May 1917, p. 302	$1.17(1+3.6/d+0.03d)$	$0.495(1+3.6/d+0.03d)$
Unwin...	Encycl. Brit., Vol. 12, p. 508	$1.4(1+3.6/d)$	$0.475(1+3.6/d)$

In the average power plant where the pipe lines are comparatively short, no attempt is ordinarily made to calculate the pressure drops through the pipe itself, and the diameter is proportioned on an assumed maximum-velocity basis. The pressure drops through the dry pipe, superheater, valves, and fittings for the assumed maximum velocity are obtained from the manufacturer or approximated from the results obtained in plants having similar equipment. Where very long steam lines are

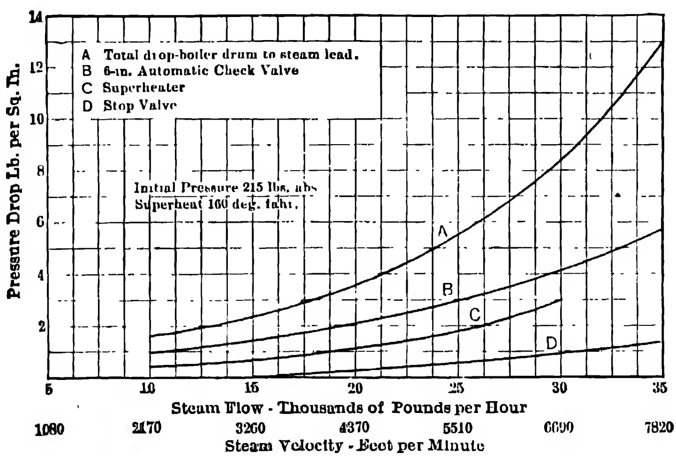


FIG. 561. Steam Pressure Drop, 500-hp. B. & W. Boilers.

employed, the friction loss in the pipe itself is considerable, and in lieu of other information it is common practice to use equations (272) to (274), preference being given to **Babcock's, Spitzglass'** and **Fritzsche's** values for coefficients C and K. Babcock's, Spitzglass' and Fritzsche's coefficients give practically the same results for moderate rates of discharge and pipe diameters under 10 in., but for larger pipe diameters and high rates of discharge Fritzsche's coefficient appears to give results more in accord with actual performance.

In the modern turbine plant where the distances between boilers and prime movers are comparatively short and the valves and fittings are designed to offer low resistance heads, a convenient rule is to allow a maximum velocity of 1000 to 1250 ft. per min. per in. of pipe diameter, the higher value for diameters over 12 in. The same rule applies to reciprocating engine plants where a large receiver is placed close to the engine throttle. Where the valves and fittings offer considerable resistance to the flow or in case of reciprocating engines without a receiver effect at the throttle, the maximum velocity is taken as 75 to 80 per cent of that given above.

The logical procedure, of course, is to proportion the pipe and fittings so that a predetermined pressure drop between the points under consideration will not be exceeded, but unfortunately, none of the rules previously mentioned can be relied upon for accurate results. In the light of the best evidence available at the present time, preference should be given to Spitzglass' equation for low-pressure steam (5 to 50 lb. abs.), saturated or wet; to Babcock's equation for dry or moderately superheated steam (100 to 200 lb. gage and pipe sizes under 10 in. in diameter); and to Fritzsche's for superheated steam flowing at high velocities, 100 ft. per sec. or more.

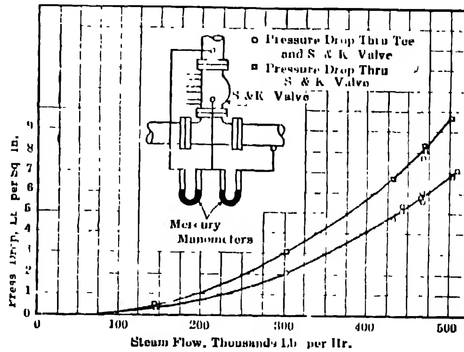


FIG. 562.

For convenience in application, equation (272a) has been transposed so that the diameter of the pipe and coefficient K have been included in a factor k, the value of which for Babcock's and Spitzglass' coefficient of friction are given in Table 100.

Equation (272a) transposed is

$$w = c\sqrt{pyd^5/L} = k\sqrt{py/L} \tag{273}$$

in which c = a coefficient involving the various reduction constants and the coefficient of frictional resistance.

in which k = a factor including c and d.

Other notations as in equation (272a).

Fritzsche's coefficient when combined with the rest of the equation reduces to the form

$$p = 0.0000516\, v^{1.85} y^{0.85}\, L/d^{1.27} \tag{273a}$$

$$p = 0.8\, w^{1.85}\, L/yd^{4.97}. \tag{274}$$

Since the weight of steam discharged through any system of piping is a function of the pressure drop, it is evident that the greater the pressure drop, the larger will be the weight discharged per unit of time. A large drop in pressure permits of a smaller pipe and, because of the reduced surface, the radiation losses will be lower, but a point is soon reached where the economy in the size of pipe is more than offset by the loss in available energy due to the reduced pressure at the point of application. There seems to be no fixed rule for determining the drop most suitable for any

given set of conditions, since the velocity factor must also be considered, particularly with wet or saturated steam, because of possible damage to fittings and joints by vibration and water hammer. In reciprocating engine practice involving the use of saturated steam and in which the pipe leads directly to the inlet nozzle, the maximum drop in pressure is ordinarily limited to 1/2 to 1 1/2 lb. per 100 ft. of straight pipe. In a number of installations in which a large receiver is placed next to the inlet nozzle, pressure drops of 1.5 to 2.5 lb. per 100 ft. of pipe have given satisfactory results. For very long pipe lines the pressure drop per 100 ft. must necessarily be small if low pressures at the point of delivery are to be avoided. In steam turbine practice involving the use of high pressure and superheat, pressure drops as high as 2.5 lb. per 100 ft. of pipe have been allowed during periods of maximum discharge. It must be remembered that the pressure drop through the pipe itself is usually but a small portion of the total drop from boiler to prime mover because of the additional resistance of the dry pipe, superheater, valves, and fittings; consequently, large pressure drops through the piping alone may cause excessive drops from boiler to prime mover unless special attention has been paid to the selection of low-resistance valves, fittings, etc.

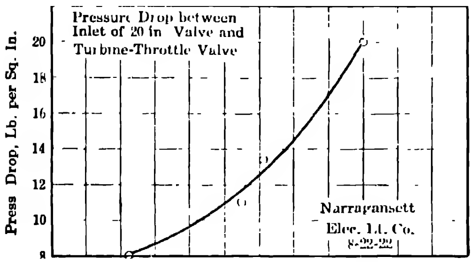


FIG. 563. Pressure Drop through 20-in. Valve and Tee. Narragansett Elec. Co.

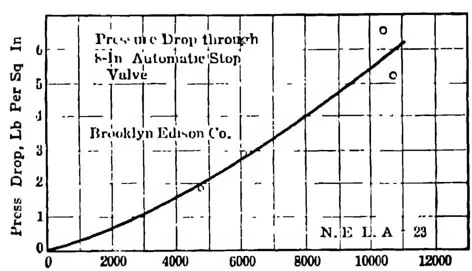


FIG. 564.

The average pressure drop in exhaust steam mains varies from 0.2 to 0.4 lb. per 100 ft. for non-condensing service and from 0.2 to 0.4 in. of mercury per 100 ft. for a vacuum of 26 in. In large steam turbine installations there is practically no exhaust piping and steam velocities of 300–400 ft. per sec. are possible with a negligible pressure drop.

Attempts to include factors for condensation or radiation losses merely complicate the problem without adding to its accuracy. The losses must be considered, of course, in estimating the probable condition of the steam at the end of the line, but except for bare pipe and very long covered lines

the influence of the heat loss on the density of the steam is less than the inaccuracies in the accepted formulas themselves.

The application of Babcock's, Spitzglass' and Fritzsche's equations is best illustrated by the following examples:

Example 86. — Calculate the size of pipe necessary to deliver 36,000 lb. of saturated steam per hr. through a covered pipe 1000 ft. long if the pressure drop is to be approximately 5 lb. per sq. in. Initial abs. pressure 125 lb. per sq. in.

Solution. — Here $w = 36{,}000 \div 3600 = 10$; $p = 5$; $L = 1000$; mean value of $y = 0.2737$.

Substituting these values in equation (273) and solving:

$$10 = k\sqrt{5 \times 0.2737 \div 1000}, \text{ or } k = 270.$$

From Table 100 we find that this value of k corresponds to an internal diameter of 8.7 in. (by interpolation) for Babcock's equation and 9 in. for Spitzglass'.

Fritzsche's equation (274) gives

$$5 = (0.8 \times 10^{1.85} \times 1000) \div (0.2737\, d^{4.97}), \text{ or } d = 8.49 \text{ in.}$$

"Standard" wrought-steel pipe is suitable for a pressure of 125 lb., and since the inside diameter of a 9-in. pipe is 8.94, the nearest to the calculated values, it is evident that the different equations give results substantially in agreement for the given conditions, namely, a standard weight 9-in. pipe.

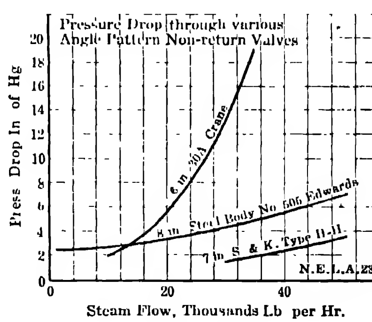


Fig. 565.

Example 87. — Calculate the size of pipe necessary to deliver 432,000 lb. of steam per hr. through a turbine lead 150 ft. long, if the pressure drop in the pipe is to be approximately 3 lb. per sq. in. Initial abs. pressure 350 lb. per sq. in. and superheat 300 deg. fahr.

Solution. — Here $w = 432{,}000 \div 3600 = 120$; $p = 3$; $L = 150$; $y = 0.495$ (the change in density due to the heat drop in this length of properly covered pipe is negligible).

Substituting these values in equation (273) and solving

$$120 = k\sqrt{3 \times 0.495 \div 150}, \text{ or } k = 1206.$$

From Table 100 we find that this value of k corresponds to an internal diameter of 15.4 in. for Babcock's equation and 16.6 for Spitzglass'.

Fritzsche's equation (274) gives

$$3 = (0.8 \times 120^{1.85} \times 150) \div (0.495 \times d^{4.97}), \text{ or } d = 14.3.$$

From Table 90 it will be seen that O.D. pipe with 5/8-in. thickness of wall is specified for 350 lb. pressure and 15 to 16 in. outside diameter.

Babcock's rule, therefore, calls for 15.4 + 1.25 = 16.65 O.D. Either a 16-in. or 17-in. pipe may be selected, depending upon the allowable deviation from the specified pressure drop. Spitzglass' rule calls for an 18-in. O.D. pipe and Fritzsche's for a 15-in. or a 16-in. O.D. pipe.

The higher the velocity and the larger the diameter, the greater will be the variation in results based upon these equations. For example, the calculated pressure drops per 100 ft. in an 18-in. inside diameter pipe through which steam at an absolute pressure of 500 lb. and temperature of 767 deg. fahr. is flowing at a velocity of 300 ft. per sec. are as follows: Spitzglass, 10.3 lb. per sq. in.; Babcock, 7.57 lb. per sq. in.; Fritzsche, 4.39 lb. per sq. in.

TABLE 100

VALUES OF k FOR VARIOUS PIPE DIAMETERS

Inside Diameter, In.	k		Inside Diameter, In.	k	
	Babcock	Spitzglass		Babcock	Spitzglass
0 5	0 028	0 028	8	217	198
0 75	0 297	0 287	9	298	268
1.0	0 675	0.66	10	392	348
1.5	2 16	2.34	11	505	446
2.0	4 90	4 75	12	635	550
2.5	9 20	8 85	13	780	674
3.0	15 2	14 6	14	950	800
3.5	23.2	22.3	15	1138	955
4 0	33 8	32 3	16	1342	1108
4.5	46 3	43 9	17	1580	1270
5.0	62 1	58 0	18	1730	1480
5 5	79.5	74 6	19	2100	1680
6.0	99.5	94 0	20	2390	1905
6 5	125	115	21	2700	2138
7 0	152	142	22	3060	2400
7 5	182	159	24	3840	2930

Numerous experiments have been conducted on valves and fittings with a view of formulating some rule or set of rules by means of which the pressure drops may be calculated for different rates of discharge and for varying steam conditions, but the results have been far from harmonious. The pressure drop is expressed either (1) directly in lb. per sq. in., or the equivalent, or (2) as an added length of straight pipe equivalent in resistance to the various valves, fittings and bends. Scattering tests on various types of valves and fittings show that the friction drop follows the law

$$p = Cv^n/d^m \tag{275}$$

in which

p = pressure loss, lb. per sq. in.,

C = a coefficient depending upon the shape of the fitting and including certain reduction constants,

v = velocity, ft. per sec.,
m, n = experimentally determined exponents,
d = inside diameter, in.

Values for C, m, and n have been fairly well established for a number of small-sized standard screw fittings and valves for the flow of water, and for a few special types of valves and fittings for the flow of steam, but sufficient data are not available to render equation (275) of much service in steam piping design.

The values (see equation 278), calculated by **Foster**,[1] are perhaps as reliable as any, but since they are based on experimental data obtained from the flow of water they must be used advisedly in steam-piping calculations.

Briggs'[2] rules are frequently used in this connection, but results calculated from them are not in accord with the few scattering tests conducted under high-pressure conditions. They appear to be satisfactory for low-pressure service. According to Briggs, the length L of straight pipe in inches equivalent to the resistance of one standard screwed 90-degree elbow is

$$L = 114\,d \div (1 + 3.6/d) \tag{275a}$$

and that of a standard globe valve

$$L = 75\,d \div (1 + 3.6/d) \tag{275b}$$

Valves and fittings which offer considerable resistance to the flow of steam should be avoided in the modern power house, and special precaution is used in doing away with sharp turns and in employing valves and fittings designed for low-friction heads.

Some idea of the pressure drops in piping systems and special types of valves may be gained from the curves in Figs. 560 to 565.

Resistance of Fittings to Flow Through Pipes: Trans. A.S.M.E., Vol. 42, 1920, p. 649.

Numerous charts and tables based on the commonly used equations may be found in various publications. These charts and tables do away with the laborious calculations and furnish a means of rapidly solving problems involving the flow of steam.

Graphical Charts for Flow of Steam in Pipes. Trans. A.S.M.E., Vol. 42, p. 652, 657; "Helios," 27th Ed., p. 276–8; Mark's Handbook (1916), p. 1354.

Tables for Flow of Steam in Pipes: Kent's Handbook, 1923, p 929; Trans. A.S.M.E., Vol. 20, p. 347; "Steam" (B. & W. Co.), 1922, p. 318–20.

Power Line Loss with Superheated Steam: Power, Aug. 7, 1923, p. 233.

[1] Trans. A.S.M.E., Vol. 42, 1920, p. 648.
[2] Warming Buildings by Steam.

High Temperature and High Pressure Steam Lines: B. N. Broido, Trans. A.S.M.E., Vol. 44, 1922, p. 1199.

Critical Velocity in One-pipe Heating Systems: Power Plant Engrg., Sept. 15, 1922, p. 914.

Capacities of Steam Heating Risers: Jour. A.S.H. & V.E., Mar. 1923, p. 115.

309. Flow of Water through Orifices, Nozzles, and Pipes. — The flow of water through orifices, etc., has been treated so thoroughly in numerous books on hydraulics that no attempt will be made to develop the various formulas used in this connection, and only those occasionally required for power house calculations will be given.

Free discharge from orifices and nozzles. — The rate of discharge through all shapes of orifices and nozzles may be calculated with sufficient accuracy for general service from the following equation:

$$Q = CA\sqrt{2gh} \tag{275c}$$

in which

Q = rate of discharge, cu. ft. per sec.

C = an experimentally determined coefficient varying with the head and shape of opening.

A = area of the orifice, sq. ft.

h = head of water producing flow, ft.

g = acceleration of gravity = 32.2.

For circular orifices with sharp edges

$$C = 0.59 \text{ to } 0.65; \text{ average } 0.60.$$

For short cylindrical nozzles 2.5 diameters in length with sharp edges

$$C = 0.81 \text{ to } 0.83; \text{ average } 0.82.$$

With rounded edges C may be increased to 0.96.

For short nozzles and conical convergent tube, sharp edges, angle 15 deg. C = 0.94.

With rounded edges C may be increased to 0.96.

Discharge from cylindrical pipe. — There are numerous formulas for the flow of water in pipe, but it is difficult to select the one best suited for the ordinary pipes of engineering practice because there is no standard of interior roughness and the interior surface does not remain constant in service. Many of the more exact equations are complicated and require considerable time for evaluation and, unless accompanied by curves or tables giving reduction factors, are too unwieldy for everyday practice. Pipe tables, similar in purpose to steam tables, are to be found in most engineering handbooks and in practically all catalogues of pump manufacturers, so that the use of formulas may be dispensed with entirely.

Spitzglass' equation appears to check substantially with experimental data, and in connection with a table of constants for various pipe sizes, offers a simple means of calculating the flow of water in pipes. Spitzglass' equation for the flow of water in iron pipes of average smoothness and cleanliness is

$$Q = 53.4 \sqrt{d^5/(1 + 3.6/d + 0.03\,d)}\,\sqrt{h/L} \qquad (276)$$

$$= k\sqrt{h/L} \qquad (276a)$$

in which

Q = rate of flow, gal. per min.,
d = internal pipe diameter, in.,
h = friction head or pressure drop, ft. of water,
L = length of pipe, ft.,
k = a factor, including constant 53.4, for various pipe diameters.

Spitzglass' values for k are given in Table 101.

Loss of head due to friction in pipes. — All of the more exact rules for the friction head in pipes are of the form

$$h_f = CLv^m/d^n \qquad (277)$$

in which

h_f = friction head, ft. of water,
C = coefficient, including the various reduction constants,
m, n = experimentally determined coefficients.

Other notations as in equation (276).

Conrad Meir's values for C, m, and n are 0.0085, 1.86 and 1.25 respectively.

For clean commercial steel pipe and average water, the following simplified modification of equation (277) gives reasonably accurate results.

$$h_f = .0045\,Lv^2/d. \qquad (277a)$$

Wm. Cox (Am. Mach., Dec. 28, 1893) gives the following empirical rule which checks up fairly well with test results.

$$h_f = L(4\,v^2 + 5\,v - 2)/1100\,d. \qquad (277b)$$

Example 88. — 200 gal. of water per min. are to be discharged through a 4-in. steel pipe line 400 ft. long. Calculate the pressure drop by the various rules given above.

Solution. — $v = (200 \times 144) \div (7.48 \times 60 \times 12.72) = 5$ ft. per sec. $d = 4$; $L = 400$; $k = 1220$ from Table 101.

TABLE 101

VALUES OF COEFFICIENT k

(Spitzglass)

Internal Diameter, In.	k	Internal Diameter, In.	k	Internal Diameter, In.	k
0.5	3.3	5.0	2,350	13	25,100
0.75	10.7	5.5	2,810	14	27,500
1.0	24.8	6.0	3,520	15	35,800
1.5	79.5	6.5	4,320	16	41,800
2.0	178.0	7.0	5,280	17	48,000
2.5	334.0	8.0	7,400	18	53,300
3.0	535.0	9.0	10,000	19	67,600
3.5	836.0	10.0	13,200	20	71,400
4.0	1220.0	11	16,650	22	90,500
4.5	1650.0	12	20,700	24	110,000

Spitzglass:

$200 = 1220 \sqrt{h_f/400}$ $\qquad h_f = 10.7$ ft.

Meir:

$h_f = 0.0085 \times 400 \times 5^{1.86}/4^{1.25}$ $\qquad h_f = 11.9$ ft.

Equation (277a)

$h_f = 0.0045 \times 400 \times 25/4$ $\qquad h_f = 11.2$

Cox:

$h_f = [(4 \times 5^2) + (5 \times 5) - 2]\ 400 \quad 1100 \times 4$ $\qquad h_f = 11.2$

Example 89. — If 600 gal. of water per min. are to be forced through a 4-in. iron pipe line 400 ft. long, what will be the pressure at the discharge end if the initial pressure is 100 lb. gage?

Solution. — Here $d = 4$; $L = 400$; $k = 1220$; $Q = 600$; 2.3 ft. of water = 1 lb. per sq. in. Let p_2 = final pressure, lb. per sq. in., then $h_f = 2.30\ (100 - p_2)$. Substituting these values in equation (276a) and solving

$$600 = 1220 \sqrt{2.3\ (100 - p_2)/400}, \quad p_2 = 58 \text{ lb. per sq. in.}$$

Loss of head due to friction of fittings. — The law for the friction loss through fittings is, according to the latest experiments, of the same general form as equation (277) except that coefficient C and length L are combined to form a single experimentally determined coefficient, K. According to Foster, the friction drops for the flow of water through various standard screw fittings may be calculated from the formula

$$L = 2.47\ rd^{1.25} \tag{278}$$

in which

L = equivalent length of standard pipe to allow for the fitting under consideration, ft.

r = an experimentally determined resistance factor,
d = inside diameter of pipe to which fitting is attached, in.

Foster[1] assigns the following values for r: gate valve, 0.25, long-sweep elbow or one run of standard tee, 0.33; standard 90 deg. elbow, 0.42; angle valve, 0.90; close return bend, 1.00; globe valve, 2.00. For steam use 2.21 in place of 2.65 in equation (278).

A rough rule is to assume that the friction head, ft. of water, varies with the square of the velocity, thus

$$h_f = Cv^2/2\,g \tag{278a}$$

C having the following values

	Angles			*Class of Valve*	
	45 deg.	90 deg.	Gate	Globe	Angle
C	0.182	0.98	0.182	1.91	2.91

Because of the great variation in design of valves and fittings there is naturally a wide range in the values of the experimentally determined coefficients, and the constants given above must be used advisedly. For data pertaining to special experimental research consult the accompanying bibliography.

Flow of Water in Short Pipes: Trans. A.S.M.E., Vol. 45, 1923.

Flow of Water through One and One-half Inch Pipes and Valves: Purdue, Engrg. Exp. Station, Bul. No. 1, 1918.

Hydraulic Experiments with Valves, Orifices, Hose, Nozzles, and Orifice Buckets: Univ. Ill., Engrg. Exp. Station, Bul. No. 105, 1918.

Average power plant practice gives the following maximum velocities of water flow in clean iron pipes.

Size of Pipe in Inches	Velocity, Ft. per Minute	Size of Pipe in Inches	Velocity, Ft. per Minute
$\frac{1}{8}$ to $\frac{1}{2}$	50–100	3 to 6	300–500
$\frac{1}{2}$ to $1\frac{1}{2}$	100–200	Over 6	500–800
$1\frac{1}{2}$ to 3	200–300		

Friction through Condenser Tubes. — The following equation is commonly used in this connection

$$h_f = C_v^{1.83}\,L + 2\,N \tag{279}$$

in which

N = No. of passes; other notations as previously given.

C = 0.016 for 5/8-in. tubes; 0.012 for 3/4-in.; 0.008 for 1-in. for clean tubes. Add 20 per cent for dirty tubes.

[1] Trans. A.S.M.E., Vol. 42, 1920, p. 649.

Example 90. — Calculate the pressure loss in a 2-pass surface condenser having 1-in. tubes 20 ft. long if the velocity of flow is 6 ft. per sec.

Solution. — Substitute $C = 0.008$, $L = 20$ and $v = 6$ in equation (279) and solve, thus:

$$h_f = 0.008 + 6^{1.83} \times 20 + 2 \times 2 = 8 \text{ ft.}$$

310. Stop Valves — Hand Operated. — The valves used to control and regulate the flow of fluids are the most important element in any piping system. A good valve should have sufficient weight of metal to prevent distortion under varying temperature and pressure, or under strains due to connection with the piping; the seats should be easily repaired or renewed; there should be no pockets or projections for the accumulation of condensation dirt and scale, and the valve stem should permit of easy and efficient packing. Stop valves are made in such a variety of designs that a brief description will be given of only a few fundamental types.

Figure 566 shows a section of an ordinary **globe valve,** so called because of the globular form of the casing. This type of valve is the most common

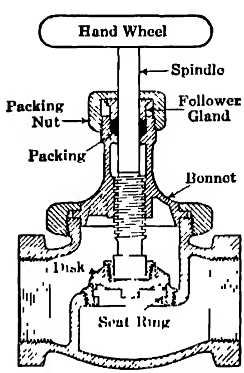


FIG. 566. Typical Globe Valve, Screw-top, Inside-screw.

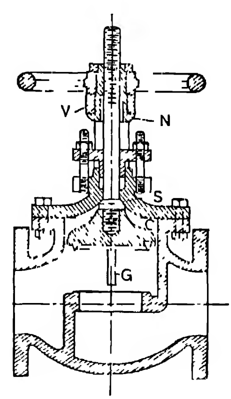


FIG. 567. Typical Globe Valve, Bolt-top, Outside-screw.

in use for small sizes. Globe valves are designated as (1) **inside screw** and (2) **outside screw,** according as the screw portion of the stem is inside the casting, Fig. 566, or outside, Fig. 567. The top, or bonnet, may be screwed into the body of the valve, Fig. 566, or bolted, Fig. 567. The smaller sizes, 3 in. and under, are usually of the **screw-top** type and the larger of the **bolt-top** type. Valves with **outside yoke** and **screw** are preferable to others, in that they show at a glance whether the valve is open or closed, an advantage in changing from one section to another. The discs are made in a variety of forms, the material depending upon

the nature of the fluid to be controlled. Thus, for cold water, hard rubber composition gives good results; for hot water and low-pressure steam, Babbitt metal; for high-pressure saturated steam, copper or bronze; and for highly superheated steam, monel metal. The valve bodies are of brass for low-temperature sizes under 3 in., cast iron for the larger sizes and ordinary pressures and temperatures, and cast steel or forged steel for high temperatures and pressures. Globe valves should always be set to close against the pressure, otherwise they could not be opened if the valves should become detached from the stem. Globe valves should never be placed in a horizontal steam return pipe with the stem vertical, because the condensation will fill the pipe about half full before it can flow through the valve. Globe valves that are open all the time are preferably designed with a **self-packing spindle,** as in Fig. 567, in which the top of shoulder C can be drawn tightly against the under surface of bonnet S, thus preventing steam from leaking past the screw threads while the spindle is being packed. For low pressures such as are encountered in heating service, a **packless valve** of the type illustrated in Fig. 568 is finding favor with engineers. The sylphon bellows encloses

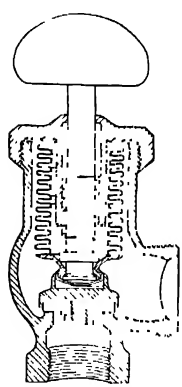


Fig. 568. Typical Packles Valve.

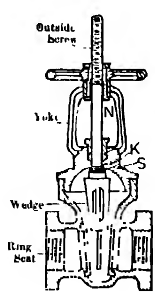


Fig. 569. Typical Low-pressure Gate Valve, Outside-screw and Yoke.

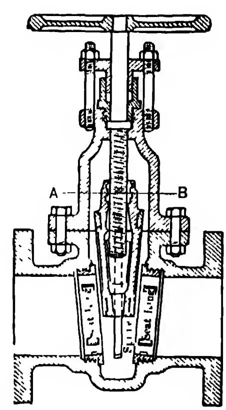


Fig. 570. Typical Gate Valve, Solid-wedge, Bolt-top, Inside-screw.

the valve stem so that the stuffing box is dispensed with. Small valves for high-pressure superheater drains are frequently designed of forged steel.

Figures 569 to 573 show different designs of **gate** or **straightway** valves. These valves offer little resistance to the flow of fluid passing through them and are designed with the same range in body design and materials as the globe valves. Figure 570 shows a section through a typical cast-

iron, bronze-trimmed valve with **solid-wedge** gate and suitable for moderate pressures and temperatures. For the sake of illustration this valve is fitted with inside screw. In this design the spindle remains stationary so far as any vertical movement is concerned and the gate or plug, being attached to it by means of a threaded unit, rises into the bonnet when the spindle is revolved. It is impossible to tell by its appearance whether a valve of this form is open or closed. Valves with inside screw are adapted to situations where there is considerable external dirt and grit, since the screw is enclosed and protected.

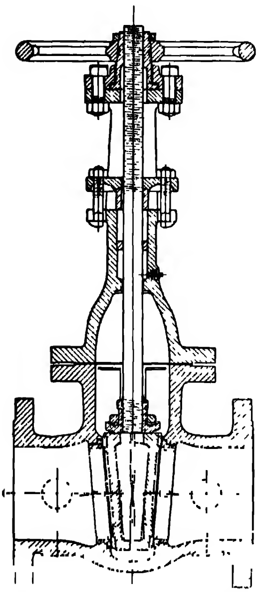


Fig. 571. Typical High-Pressure Gate Valve, Split-wedge.

Figure 571 shows a section through a cast-steel valve with **split-wedge** gate and monel trimmings, designed for high pressures and temperatures. This particular design is fitted with outside screw and yoke. This construction is a perfect indicator to show whether the valve is open or shut, as the hand wheel is stationary and the spindle rises in direct proportion to the amount the valve is opened. Practically all power stations using high-pressure superheated steam have standardized on the steel-body gate valve with complete chrome nickel trimmings. All high-pressure valves above 8 in. in diameter should be provided with a small by-pass valve, as the pressure exerted against the disc or gate is very great when the valve is closed, and the force required to move it is considerable. The by-pass valve also facilitates "warming up" the section to be cut in and is more readily operated than the main valve.

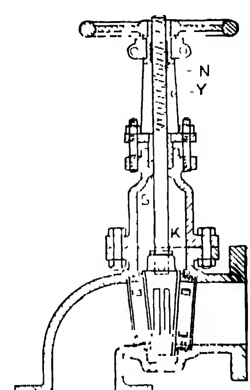


Fig. 572. Typical Angle Valve, Gate-pattern.

311. Stop Valves — Remote Control. — In the modern steam power plant, steam-header and sectionalizing valves and the large valves in the condenser circulating-water pipe line are usually power operated. This greatly reduces the time of opening and closing the large valves and permits of remote control. If a bad break should occur in the high-pressure steam line, it would be almost impossible to locate it and sectionalize the header by hand-controlled valves, on account of the tremendous volume

of steam escaping and the consequent confusion. With power-operated valves this opening and closing can be effected at any distant point by the use of a suitable control system. Both hydraulically, and electrically operated gate and globe valves for all purposes are on the market.

Figure 573 shows a section through a typical hydraulically controlled gate valve, and Fig. 574 gives a diagrammatic outline of the control system at the Northeast Station of the Kansas City Power & Light Co. All of the valves on the circulating water lines in the condenser well, and also those on the steam headers in the boiler room, are provided with hydraulic cylinders. Oil is used as the operating fluid and is supplied at a pressure of 150 lb. gage. The pump is automatically started and stopped and the pressure on the system is maintained uniform by means of a weighted accumulator. The two ends of the hydraulic cylinder on each valve communicate with the oil system through a four-way plug cock which can be placed at any convenient point. Turning the cock admits oil to either side of the piston and exhausts it from the other, and this in turn opens or closes the gate. Admission of oil to the valves causes the accumulator to descend to such a point that the float switch closes and the pump is started. The pump continues to operate until the demand for oil is over and the counterweight, attached by a cable to the accumulator, reaches its extreme position and opens the circuit.

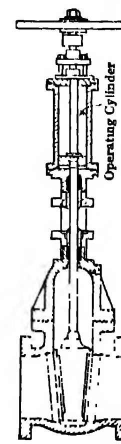


FIG. 573. Typical Hydraulically Operated Gate Valve.

Figure 575 shows the general details of an electrically operated high-

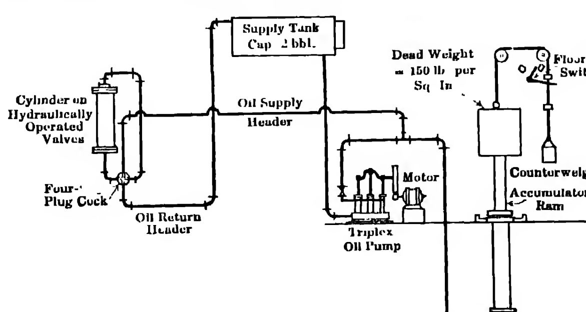


FIG. 574. Hydraulically Operated Valve System.

pressure steam globe valve provided with a declutching device to insure automatic stopping without jamming when the motor armature and gears "drift" after tripping. It is also equipped with a self-contained electrical

limit capable of breaking the main motor current without arcing, and so geared that operation of the valve by hand will not throw it out of adjustment. To open or close the valve, it is only necessary to manipulate a switch or push button placed at any convenient point. When such a valve is used for high-pressure steam service, there are usually three points of control: (1) a local control station mounted at some point from which opening and closing the valve may be observed, (2) a remote control in the boiler room close to the door leading into turbine room and (3) a remote control at a safe point unlikely to be affected by steam flow. In some plants, a centralized control board is adopted. Figure 576 shows the location of the electrically operated valves in the Hell Gate Station of the United Electric Light & Power Co.

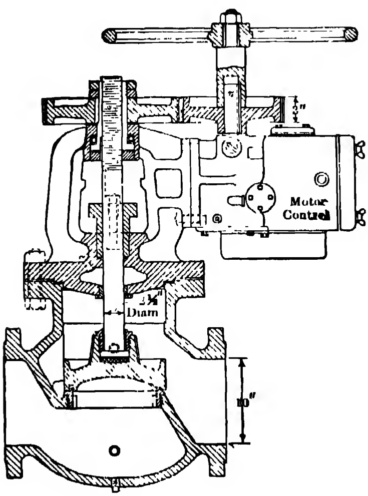


FIG. 575. Dean Electrically Controlled Valve.

In steam heating plants, the supply valves on the heating coils are frequently controlled by a thermostat. Each valve may be controlled by a direct-acting individual thermostat, or by a thermostat of the relay type which controls the motive power actuating the valve. The direct acting controls are usually of the sylphon type in which a small temperature variation effects a considerable change in length of the bellows. This change in length opens and closes a balanced stop valve.

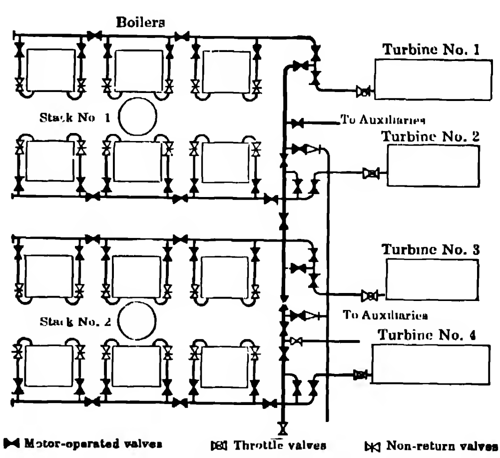


FIG. 576. Location of Electrically Operated Valves, Hell Gate Station.

Figure 577 shows a section through a **Powers thermostat** illustrating a relay type of mechanism for regulating the supply of compressed air to a stop valve of the diaphragm type.

The expansible disc U contains a volatile liquid having a boiling point of about 50 deg. fahr. The pressure of the vapor within the disc at a temperature of 70 deg. amounts to 6 lb. to the sq. in., and varies with every change of temperature, causing a variation in the thickness of the disc. The disc is attached by a single screw O to the lever Q, which rests upon the screw F as a fulcrum. The flat spring R holds the lever and disc against the movable flange M. Connecting with the chamber N are two air passages H and I. The thermostat is attached by means of two screws at the upper end to a wall plate permanently secured to the wall. This wall plate has ports registering with H and I, one for supplying air under pressure and the other for conducting it to the diaphragm motor which operates the valve or damper. Air is admitted through H under a pressure of about 15 lb. per sq. in., and its passage into chamber N is regulated by the valve J, which is normally held to its seat by a coil spring under cap P. K is an elastic diaphragm carrying the flange M, with escape valve passage covered by the point of valve L. Valve L tends to remain open by reason of the spring. When the temperature rises sufficiently, expansion of the disc U first causes the valve to seat, its spring being weaker than that above valve J. If the expansive motion is continued, valve J is lifted from its seat and compressed air flows into chamber N, exerting a pressure upon the elastic diaphragm K in opposition to the expansive force of the disc. If the temperature falls, the disc contracts and the overbalancing air pressure in N results in a reverse movement of the flange M, permitting the escape valve to open and discharge a portion of the air; thus the air pressure is maintained always in direct proportion to the expansive power (and temperature) of the disc U. The passage I communicates with a diaphragm valve, Fig. 578. The compressed air operates the diaphragm against a coiled spring resistance, so that the movement is proportional to the air pressure and the supply of steam is controlled accordingly. The adjusting screw G, squared to receive a key, carries an indicator by means of which the thermostat can be set to carry

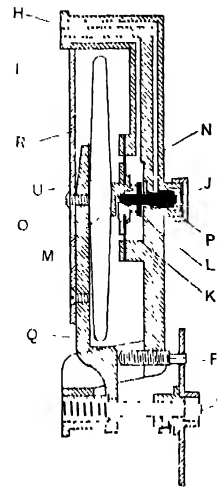


FIG. 577. Powers Thermostat

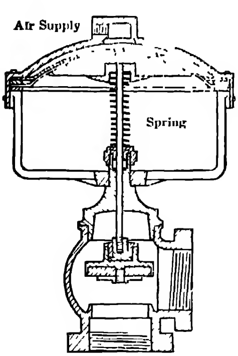


FIG. 578. Typical Diaphragm Valve.

any desired temperature within its range, usually from 60 to 80 deg. In changing the temperature adjustment, lever Q forces the disc U closer to or farther from the flange M.

In connecting up the system, compressed air is carried to the thermostat and diaphragm valves, from a reservoir, through small concealed pipes.

In the indirect system of heating, the dampers are of the diaphragm type and the method of regulation is the same as with the direct system.

Sectionalizing and Remote Control of High-pressure Steam Lines: Mech. Engrg., Aug., 1923, p. 483.

Electrically Operated Valves: Power Plant Engrg., June 1, 1923, p. 572.

312. Emergency Closing Valves. — In addition to the remote-control power-actuated stop valves which can be quickly closed in case of emergency, there are a number of valves on the market intended primarily for emergency service. One of the simplest of these is the **butterfly valve,** which is similar in principle to the weighted check illustrated in Fig. 581D. The disc is held open by a trigger device which may be operated manually by a cord or electrical push button, or automatically by any pressure or

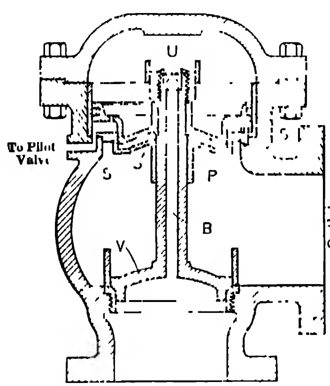


FIG. 579. Typical Triple-acting Emergency Valve.

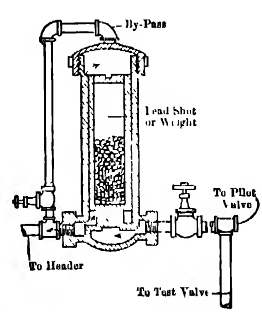


FIG. 579a. Pilot Mechanism for Emergency Valve

speed variation. Releasing this trigger causes the weight attached to the disc-shaft lever to drop, which in turn quickly closes the valve. A by-pass is provided so that the pressure on both sides of the disc can be equalized when the valve is restored to its open position. Valves of this type are commonly placed on turbine and engine leads and the releasing mechanism is arranged so that it may be tripped automatically, when the unit overspeeds, or manually from some distant point.

Figure 579 shows a section through the valve body and Fig. 579a a similar view of the pilot mechanism of a **triple-acting emergency valve,** so

called because it is intended to act as a non-return valve as well as to cut off the flow when a predetermined pressure drop occurs on either side of the valve. Referring to Fig. 579, valve disc V and piston P are secured to the same stem so that the latter rises with the lift. Piston P acts as a dashpot, a double cushioning effect being established by the confined spaces S and S'. Steam from the lower side of the valve disc passes through by-pass B into the upper chamber U and through opening O into annular space S. When the pressure on both sides of the disc is the same and the steam in space S is confined so that it cannot escape, the system is balanced and the valve is free to move. Annular space S is piped to the pilot valve. In case the pressure on the lower side of the disc is suddenly lowered, as in case of a tube blow-out, the disc will close because of the reversed flow. On the other hand, in case the pressure on the upper side of the disc should drop below a predetermined amount, the pilot valve, which is connected to this side of the line, will release the pressure in annular space S, and the excess pressure in chamber U will force the valve to its seat. The steam pressure acting on the lower side of the pilot valve may be automatically released by the lifting of the pilot-valve disc (as when the pressure in the small lead from pilot valve to header drops), or an electrically operated trip may open the pipe to the atmosphere. In case of a large and sudden drop on the upper side of the disc, the kinetic energy of the steam acting on the bottom of the disc will tend to retard and may even overcome the differential pressure acting on top of the piston.

313. Non-return or Stop-check Valves. — Where there are two or more boilers connected to a common steam header, each boiler should be provided with an automatic return valve to prevent reversal of flow. To be successful, such a valve should not open until the pressure in the boiler is equal to or slightly greater than that in the header; it should neither stick and become inoperative nor chatter and hammer while performing its duty. Figure 580 shows a section through a typical **automatic non-return valve,** which, as will be seen from the illustration, is essentially a cushioned check valve with a detached valve stem for securing the valve to its seat in case the valve is to be held closed. In some designs the cushioning effect is produced by an exterior spring connected through suitable linkage to the disc instead of an interior dashpot as illustrated in Fig. 580. In the event of a tube "blowing out" in a boiler to which a non-return valve is connected, the valve will instantly and automatically

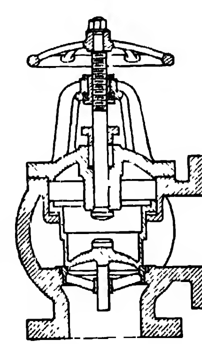


FIG. 580. Typical Automatic Non-return Valve.

close, cutting out the boiler and preventing a back flow of steam from the main. It will also act as a safety stop valve, preventing steam from being turned into a cold boiler while men are working inside, because it cannot be opened when there is pressure on the header side only.

Electrically operated non-return valves are also in use. The operation is the same as that required on triple-duty design (see paragraph 311) with the exception that the valve stem is driven by a motor with reduction gearing.

For a description of a series of low-pressure-loss non-return valves, consult Report of Prime Movers Committee, N.E.L.A., 1923, Part A, p. 42.

314. Check Valves. — Figure 581, *A* to *D*, illustrates the different types of check valves in most common use on water lines. *A* is a **ball check,** *B* a **cup** or **disc check,** *C* a **swing check,** and *D* a **weighted check.** Occasionally the valve body is fitted with a valve stem and handle for

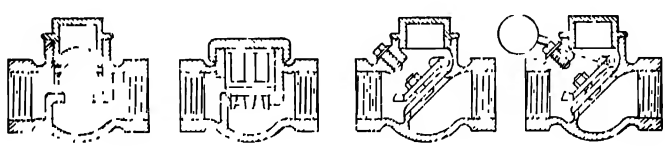


Fig. 581. Types of Check Valves.

holding the disc against its seat, in which case it is designated as a **stop check.** In *A* and *B*, the valve seat is parallel to the direction of flow and the valve is held in place by its own weight and by the pressure of the fluid in case of reverse flow. In the swing check, the seat is at an angle of about 45 deg. to the direction of flow. The latter construction is preferred as it offers less resistance to flow and there is less tendency for impurities to lodge on the valve seat. By extending the hinge of the swing through the body of the valve, a lever and weight may be attached as in *D*, and the check will not open except at a pressure corresponding to the resistance of the weight. It thus acts as a relief valve and at the same time prevents a reversal of flow. **Stop checks** are usually inserted in boiler feed lines close to the boiler, and, when locked, act as any ordinary stop valve and permit the piping to be dismantled or the regulating valve to be reground without lowering the pressure on the boiler. Since the wear on check valves is excessive and necessitates frequent regrinding, they are often mounted with **regrinding discs,** Fig. 581 (*C*), which may be " ground " against the seat without removing the valve from the line.

315. Blow-off Cocks and Valves. — The requirements of a good blow-off valve are that it shall furnish a free passage for scale and sediment, that it shall close tightly so as not to leak, and that it shall open easily without sticking or cutting. On account of the rather severe service to which such valves are subjected, they should be made very heavy, with renewable wearing parts.

Figure 582 gives a sectional view of a Crane blow-off valve suitable for 300 lb. pressure. The body and bonnet are of cast steel and the seat of

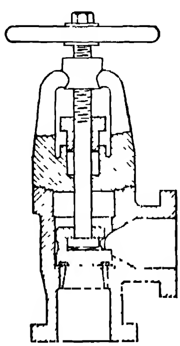


Fig. 582. Crane Blow-off Valve.

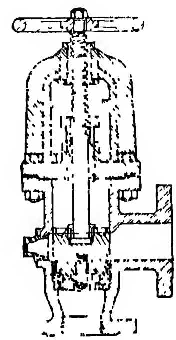


Fig. 583. Lunkenheimer Blow-off Valve.

monel metal. The disc is designed to form a throttling lip with the body so that scale cannot lodge between the seat and disc. Figure 583 shows a section through a Lunkenheimer blow-off valve suitable for 400 lb. pressure. The body and bonnet are of cast steel and the mountings of monel metal. The effect of wire drawing and the consequent rapid erosion of the seating surfaces are minimized by the piston-shaped disc which, as the valve is closed, fits concentrically within the cylinder above the seat.

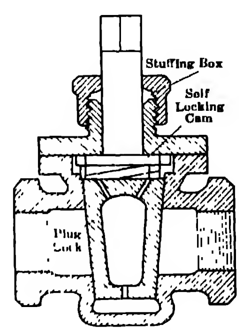


Fig. 584. Typical Blow-off Cock.

Figure 584 shows a section through a typical blow-off cock of the straightway taper-plug pattern with self-locking cam. Plug cocks are sometimes used for throttling, but they are not suitable for this service and should be used only as a protection against leakage of the blow-off valve.

Every blow-off outlet of each boiler in a battery should be equipped with a blow-off cock or a *Y* blow-off valve between the boiler and the blow-off valve, as shown in Fig. 585. When a boiler is blown off, the cock or *Y* valve should be opened first and the blowing-off operation controlled by the blow-off

valve. After blowing, the blow-off valve should be closed first, and then the cock or *Y* valve.

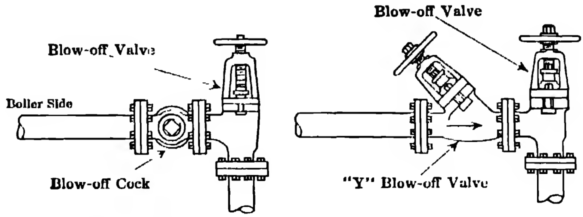


FIG. 585. Arrangement of Blow-off Valves and Cocks.

316. Safety Valves. — The **dead weight** is the simplest form of safety valve. The valve is held on its seat against the boiler pressure by a cast-iron weight. This type has the advantage of great simplicity, and can be least affected by tampering, since it requires so much weight that any additional amount which would seriously overload it can be quickly detected.

In the **lever-type** of safety valve, the valve is held against its seat by a loaded lever, thereby permitting the use of a much smaller weight than the "dead-weight" type, since the resistance is multiplied by the ratio of the long arm of the lever to the short one. The proper position of the weight is determined by simple proportion. The use of safety valves of the "dead-weight" or "lever" type for high-pressure service is prohibited in U. S. marine service and in most states and should be completely discontinued since these valves are not only unreliable but possess many operating disadvantages when compared with the spring-loaded device.

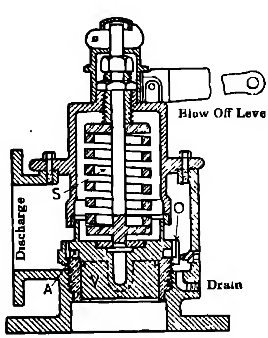


FIG. 586. Typical "Pop" Safety Valve.

Figure 586 shows a section through a typical **pop safety valve** in which the boiler pressure is resisted by a spring. This type of valve has practically supplanted all other forms. The boiler pressure acting upon the under side of valve *V* is resisted by the tension in spring *S*. As soon as the boiler pressure exceeds the resistance of the spring, the valve lifts from its seat and the steam escapes through opening *O*. The static pressure of the steam plus the force of its reaction in being deflected from the surface *A* holds the valve open until the pressure in the boiler drops about 5 lb. below that at which the valve is lifted. The additional area of valve exposed to pressure when the valve lifts causes it to open with a sudden motion

which has given it its name, and it also closes suddenly when the pressure has fallen. These valves are arranged so that the spring tension may be varied without taking them apart, and provision is made for lifting the seats by means of a lever. The seats are of solid nickel or monel metal in the best designs, to minimize corrosion.

The commercial rating of a safety valve is based upon the area exposed to pressure when the valve is closed.

The number and size of safety valves for a given boiler are ordinarily specified by insurance, city, or state legislation.

The logical method for determining the size of safety valves is to make the actual opening at discharge sufficient to take care of all steam generated at maximum load without allowing the pressure to rise more than 6 per cent above the maximum allowable working pressure, thus:

Let W = maximum weight of steam discharged, lb. per hr.,
A = effective discharge area, sq. in.,
P = boiler pressure, lb. per sq. in., abs.,
L = lift of valve, in.,
K = coefficient determined by experiment,
D = diameter of valve, in.

According to **Napier's rule** for the discharge of steam through unrestricted orifices

$$W = 3600\ PA/70 = 51.4\ PA. \tag{280}$$

Allowing 0.96 for restriction of orifice (A.S.M.E. Code)

$$W = 49.3\ PA. \tag{280a}$$

For a flat-seated valve, $A = \pi DL$

whence $$W = 155\ PDL \text{ and } D = 0.00645\ W/PL. \tag{280b}$$

For the almost universal 45-deg. seated valve

$$A = \pi DL \text{ sine } 45 \text{ deg.}$$

whence $$W = 109.7\ PDL \text{ and } D = 0.00911\ W/PL. \tag{280c}$$

The present rule of the United States Board of Supervising Inspectors is

$$a = 0.2074\ w/P \tag{280d}$$

in which

a = area of the safety valve in sq. in. per sq. ft. of grate surface,
w = lb. of water evaporated per sq. ft. of grate surface per hr.

This rule assumes a lift of 1/32 of the nominal diameter and 75 per cent of the flow calculated by Napier's rule. The 75 per cent corresponds nearly to the cosine of 45°, or 0.707.

Example 90a. — A boiler at the time of maximum forcing uses 2150 lb. of screened-nut Illinois coal per hr.; heat value 12,100 B.t.u. per lb.; boiler pressure 225 lb. per sq. in. gage; feedwater 200 deg. fahr. Required the size of safety valve.

Solution. — Assuming a boiler efficiency of 75 per cent, the total maximum evaporation is

$$W = 2150 \times 12{,}100 \times 0.75 \div 1033 = 18{,}880 \text{ lb. per hr.}$$

(1033 = heat content of 1 lb. of steam at 225 lb. gage above 200 deg. fahr.)

Assuming a lift of 0.1 in., we have, from equation (280*c*),

$$D = 0.0091 \times 18{,}880 \div 240 \times 0.1 = 7.17 \text{ in.}$$

According to the A.S.M.E. code, two valves would be required. Considering two valves having the same lift as the single valve, the diameter of each for the given condition would be 7.17/2 = 3.5 in. (approx.).

The following rules pertaining to safety valves are taken from the A.S.M.E. Boiler Code:

Each boiler shall have two or more safety valves, except a boiler for which one safety valve 3 in. in size or smaller is required.

One or more safety valves on every boiler shall be set at or below the maximum allowable working pressure. The remaining valves may be set within a range of 3 per cent above the maximum allowable working pressure, but the range of setting of all of the valves on a boiler shall not exceed 10 per cent of the highest pressure to which any valve is set.

Each valve shall have full-sized direct connection to the boiler. No valve of any description shall be placed between the safety valve and the boiler, nor on the discharge pipe between the safety valve and the atmosphere.

Every superheater shall have one or more safety valves near the outlet, whose discharge capacities may be included in determining the number and size of safety valves for the boiler if there are not intervening valves between the superheater safety valve and the boiler and if the discharge capacity of the safety valves on the boiler, as distinct from the superheater, is at least 75 per cent of the total valve capacity required.

The complete A.S.M.E. Boiler Code may be purchased from the American Society of Mechanical Engineers, New York City.

The How and Why of Safety Valves: **Power, Sept. 4, 1923, p. 357.**

317. Back-pressure and Atmospheric-relief Valves. — These valves are for the purpose of preventing excessive back pressure in exhaust pipes. In non-condensing plants, such valves are designated as **back-pressure valves** and in condensing plants as **atmospheric relief valves.** In the

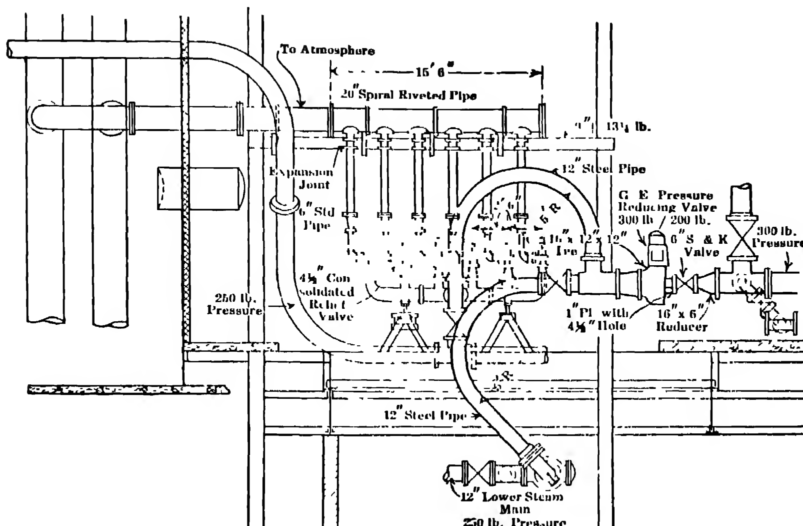


FIG. 587. Arrangement of Safety-valve Piping. L Street Station.

former, the valve is usually adjusted so that a pressure of 1 to 5 lb. above the atmosphere is necessary to lift it from its seat; in the latter, the valve lifts at about atmospheric pressure. They are practically identical in construction, differing only in minor details. A slight leakage in the back-pressure valve is of small consequence, but, in an atmospheric relief valve, it may seriously affect the degree of vacuum and throw unnecessary work upon the air pump; hence, it is customary to "water-seal" the latter. Figure 588 shows a section through a typical back-pressure valve. The valve proper consists of a single disc moving vertically. The valve stem is in the form of a piston or dashpot which prevents sudden closing or hammering. The pressure holding the valve against its seat is regulated by a spring. When the back pressure becomes greater than atmospheric plus that added by the spring, the valve rises from its seat and relieves it.

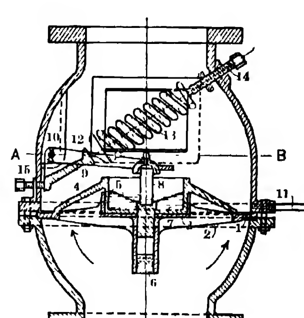


FIG. 588. Typical Back-pressure Valve. (Single-seated, Spring-loaded.)

Figure 589 shows a section through a back-pressure valve of the double-seated, lever-loaded type, in which the resisting pressure is varied by means of a lever and weight.

Figure 555 shows the application of a back-pressure valve to a typical heating system.

Figure 590 shows a section through a typical **atmospheric-relief valve.** Opening *B* is connected to the exhaust pipe and opening *A* leads to the atmosphere. Under normal conditions of operation atmospheric pressure

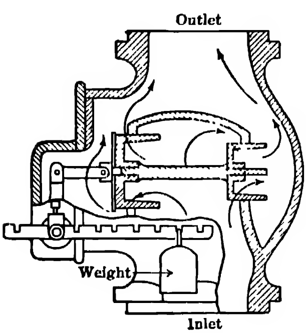


FIG. 589. Typical Back-pressure Valve. (Double-seated, Lever-loaded.)

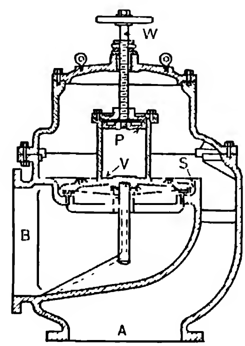


FIG. 590. Typical Atmospheric-relief Valve.

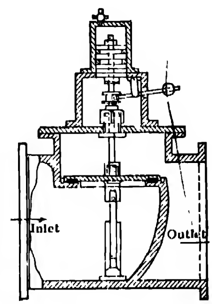


FIG. 591. Typical Atmospheric-relief Valve. Counterweight Type.

holds valve *V* against its seat. Water in groove *S* "water-seals" the seat and prevents air from being drawn into the condenser. In case the pressure in pipe *B* becomes greater than atmospheric, it lifts valve *V* from its seat and is relieved. Piston *P* acts as a dashpot and prevents the valve from slamming.

Figure 591 shows a section through an atmospheric relief valve in which the weight of the valve is counterbalanced or even overbalanced by an adjustable weight and lever, thereby permitting the valve to open at or below atmospheric pressure, as may be desired.

318. Foot Valves. — Whenever a long column of water is to be moved in either a suction or delivery pipe, it is customary to place a check valve near the lower end of the column to prevent the water from backing up when the pump reverses or shuts down. The check valve placed at the end of the suction pipe is called a **foot valve.** Any check valve may be used as a foot valve, though practice limits the choice to the disc or flap type as illustrated in Fig. 592. To prevent rubbish from destroying the action, a strainer or screen is generally incorporated with the body of the valve. *A*, Fig. 592, illustrates a **single-flap,** *B* a **multi-flap** and *C* a **disc** valve composed of a nest of small rubber valves. The single-flap are

usually made in sizes 3/4 to 6 in., the multi-flap 7 to 16 in., and the disc valve in all commercial sizes from 3/4 to 36 in. For large sizes, 16 to 36 in., the multi-disc valve is given preference, since a number of the discs may be disabled without destroying its operation.

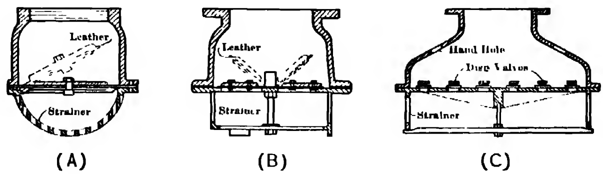


FIG. 592 Typical Foot Valves.

319. Reducing Valves. — It is frequently necessary to provide steam at different pressures in the same plant, as in case of a combined power and heating plant, or for supplying low-pressure steam to turbine glands. To effect this result, the reduction in pressure is accomplished by passing the steam through a **reducing valve,** which is but an automatically operated throttle valve.

Figure 593 shows a section through a reducing valve of the diaphragm-lever type suitable for moderate initial pressures and temperatures. The low-pressure steam acts upon the top of flexible diaphragm *D*, and the weighted lever *L* (which can be adjusted to give the desired reduction in pressure) acts upon the other side. The movement of the diaphragm causes the balanced valve *V* at the upper end of the spindle to open or close according to the variation in the low-pressure line. Inertia weights *T* and *C* prevent chattering.

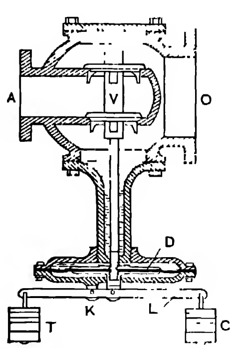


FIG. 593. Typical Reducing Valve. (Lever-loaded Diaphragm.)

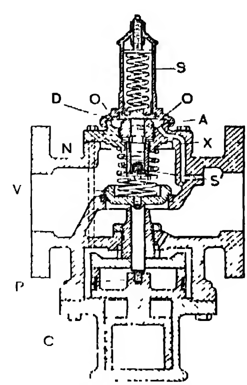


FIG. 594. Typical Reducing Valve. (Spring-loaded Diaphragm.)

Figure 594 shows a section through a reducing valve of the spring-loaded diaphragm type suitable for high initial pressures. The movement of the valve is accomplished by the reduced pressure acting through port *X*. The diaphragm is resisted by spring *S*, the tension of which may be adjusted to suit.

Auxiliary valve A is held in contact with the diaphragm by auxiliary valve spring S' and moves up and down freely with the diaphragm. As soon as auxiliary valve A is open, steam passes up around the auxiliary valve chamber through the set of holes shown under the valve seat, through the valve seat and out through the upper set of holes which communicate with port N leading to the space around the lining, passing down around the same and under piston P.

By raising piston P, main valve V is opened against the initial pressure, because the area of valve V is only one-half that of piston P; thus steam is admitted to the system.

When pressure in the system has reached the required point (determined by spring S), diaphragm D is forced upward by the reduced pressure, which passes up through port X to chamber O under the diaphragm, allowing valve A to close, thus shutting off steam from piston P. Main valve V is now forced to its seat by the initial pressure shutting off steam from the system and pushing piston P down to the bottom of its stroke. The steam beneath piston P exhausts freely around the piston (which is fitted loosely for this purpose) and passes off into the system.

In practice the main valve does not open or close entirely with each slight variation of pressure, but assumes a position which furnishes just the amount of steam required to maintain the reduced pressure desired.

Piston P is fitted with dashpot C, which prevents chattering or pounding.

Reducing valves of the types described above for reducing very high-pressure high-temperature steam to 5–10 lb. gage in a single reduction are still in an experimental stage. Two reducing valves in series with an intermediate receiver for damping fluctuations have been used successfully in this connection. A pressure-reducing valve for high pressure and temperatures, consisting of a standard globe valve actuated by the Payne Dean Control, is described in detail in the 1923 Report, Part A, p. 40, of the Committee on Prime Movers, N.E.L.A.

Reducing valves should always be by-passed to permit of repairs without shutting down the line. Care should be taken not to use too large a reducing valve, since the valve lift is very small and the larger the valve the less will be the lift for a given weight of flow and consequently the greater the wire drawing and erosion of the valve seat.

PROBLEMS

1. Determine the increase in length of a 10-in. O. D. ¾-in. thick steel pipe when cold (60 deg. fahr) and when conveying steam at 400 lb. per sq. in. gage, total temperature 750 deg. fahr

2. A 12-in. double-offset expansion U-bend having a radius of 90 in. is to take up an expansion of 1 in. Required the maximum bending stress in the bend.

3. A 10-in. O. D. $\frac{1}{4}$-in thick, steel pipe quarter bend having a radius of 60 in. is fixed at one end and free at the other. What axial force at the free end is required to deflect it 1 in. in the direction of the force.

4. Steam at 200 lb. abs. pressure is conducted through a bare standard 3-in. pipe, 500 ft. long. If the temperature of the room is 80 deg. fahr. calculate the total heat loss per hour.

5. If the pipe is covered with a single thickness of "85 per cent Magnesia" determine the saving in heat.

6. Determine the conductivity of the covering in Problem 5, per inch of thickness.

7. Determine the size of steam pipe suitable for a 10,000-kw. steam turbine using 14 lb. steam per kw-hr., initial pressure 215 lb. abs., back pressure 2 in. mercury, superheat 125 deg. fahr., if the pipe is 150 feet long and the pressure drop is not to exceed 2.0 lb. per sq. in. per 100 ft.

8. Saturated steam at 125 lb. abs. initial pressure is flowing at the rate of 20,000 lb. per hr. through a standard 6-in. pipe, 2000 ft. long. Calculate the probable pressure drop.

9. Determine the initial pressure necessary to deliver 400 gallons of water per minute through a 5-in. standard pipe 1500 ft. long, fitted with two right angle elbows and one globe valve. The water is to be discharged into an open tank.

10. How many gallons of water will be discharged through a straight length of 6-in. standard pipe 10,000 ft. long if the initial pressure is 100 lb. per sq. in., and what will be the pressure at the discharge end?

11. Determine the number and size of safety valves for a 500-hp. boiler designed to operate at a maximum load of 300 per cent above rating; boiler pressure 250 lb. abs.

CHAPTER XVII

LUBRICANTS AND LUBRICATION

320. General. — The losses due to the friction of the working parts of machinery include considerably more than the mere loss of power; namely, the depreciation resulting from the wear of bearings, guides and other wearing parts, and the expense arising from accidents traceable to defective lubrication. Perfect lubrication is one of the most essential requirements for successful operation of a plant and particularly so in the case of the turbine and other high speed machinery. The solution of the various problems of lubrication involves not only a question of the lubricant, but also methods and points of application, rate of application, and circulation if the system is a continuous one, heat dissipation, storage, and preservation. Lubrication of the various elements in a reciprocating engine plant is comparatively simple compared with that in the modern turbine installation. The operation of the steam turbine depends on a forced-feed lubrication, under pressure and at a relatively high rate of speed. As the size of the unit increases, the requirements become more and more severe and of greater importance. High peripheral speed of shaft, high unit bearing pressure, small clearance, shifting of the point of nearest approach of journal and bearing due to load changes, compression, throttling and expansion of the oil, its heating and cooling, contamination with impurities in the system, are all important factors to be particularly considered in steam turbine lubrication and lubricating systems. The lubricants most commonly met with in power plant practice are conveniently classified as oils, greases, and solids, and are usually of mineral origin, though animal and vegetable oils are occasionally used for compounding or adulterating the mineral product.

321. Oils. — *Vegetable Oils.* — Except for certain special purposes and for compounding with mineral oils, these possess lubricating properties of little practical value, since they decompose at comparatively low temperatures and have a tendency to become thick and gummy. The vegetable oils sometimes employed are linseed, cottonseed, rape, and castor.

Animal Fats. — Many animal fats have greater lubricating power than pure mineral oils of corresponding viscosity, but are objectionable on account of their unstable chemical composition. They decompose easily,

especially in the presence of heat, and set free acids which attack metals. They are seldom used in the pure state and are usually compounded with mineral oils. The animal products used in this connection are tallow, neat's-foot oil, lard, sperm, wool grease, and fish oil, the first-named being the most important. In cylinder lubrication, especially in the presence of moisture, the addition of 2 to 5 per cent of acidless tallow seems to make the oil adhere better to the metal surfaces and increases the lubricating effect, while the proportion is so small that ill effects from corrosion or gumming are scarcely perceptible.

Animal and Vegetable Oils: Power, Nov. 3, 1914, p. 636.
Compounding of Lubricating Oils: Power, Apr. 4, 1922, p. 535.
Lubrication and Lubricants: Power, Sept. 15, 1920, p. 875.
Germ Process of Lubrication: Nat. Engr., July, 1920, p. 312.

Mineral Lubricating Oils. — These are products of distillation of crude petroleum and form by far the greater part of all lubricants. They present a wider range of lubricating properties than those derived from animal or vegetable sources, the thinnest being more fluid than sperm and the thickest more viscous than fats and tallows. They are not easily oxidized, and they do not decompose, become rancid, or contain acids.

Crude Oils are grouped in three series: those of paraffin, asphaltic and cyclo-naphthene base. There is no sharp line of demarcation between these groups, since most crude oils found in all fields may contain mixtures in variable percentages of hydrocarbons belonging to all three series. Each individual hydrocarbon of any of these series has distinct physical properties, and when mixed with others the mixture frequently has properties quite different from what might be expected of the several distinct hydrocarbons which it contains. The hydrocarbons are difficult to separate and when an attempt is made to separate one compound, other hydrocarbons, both lighter and heavier, also separate from the crude. Therefore, all commercial mineral lubricants are mixtures of a number of hydrocarbons. While preference has always been given to lubricants of paraffin-crude origin, improvement in refining is placing many of the lubricants derived from asphaltic-crude oils in active competition with the former.

322. Greases. — Under this name may be included the various compounds which consist of oils and fats thickened with sufficient soap to form, at ordinary temperatures, a more or less solid grease. Those usually employed are lime, soda, or lead soaps, made with various fats and oils. "Engine" greases are thickened with a soap made from tallow or lard oil and caustic soda, and often contain neat's-foot oil, beeswax, and the like. For exceptionally heavy pressures, graphite, soapstone,

and mica are sometimes added to the grease. Table 102 gives an idea of the characteristics of a number of greases. (*Prac. Engineer, U. S.*, Apr., 1911, p. 293.) The friction tests were made on a small Thurston oil-testing machine, 320 r.p.m. and bearing pressure of 240 lb. per sq. in. of projected area. These results are purely comparative under the given conditions of rubbing surfaces, speed and pressure. For results of these greases tested on a large Olsen oil machine, consult reference given above.

TABLE 102

LUBRICATING CHARACTERISTICS OF A NUMBER OF GREASES

Type	Class	Melting Point, Deg. Fahr.	Per Cent Soap	Kind of Soap	Per Cent Free Acid as Oleic	Average Coefficient Friction
A Mineral..	Summer	167	38	Lime	Trace	0.075
B Mineral	Summer	178	20	Lime	0.3	0.054
C Mineral	Winter	165	23	Lime	6.1	0.063
D Mineral	Winter	163	16	Lime	0	0.057
E Mineral	Winter	142	19	Lime	Trace	0.046
F Tallow No. 3	Winter	125	1.4	Potash	0	0.022
G Tallow No. XX	Summer	120	2.1	Potash	0	0.029
H Lard oil		41	0			0.011

Type	Final Coefficient Friction After 3-Hr. Run	Maximum Temperature of Bearing Above that of Room, Degs. Fahr	Final Temperature of Bearing Above that of Room at End of 3-Hr. Run, Degs. Fahr
A Mineral ...	0.075	70	68
B Mineral	0.050	70	58
C Mineral ...	0.063	76	65
D Mineral	0.054	69	58
E Mineral	0.016	58	50
F Tallow No. 3	0.012	38	18
G Tallow No. XX	0.018	45	32
H Lard oil......	0.010	13	12

The following specifications cover the grade of cup grease used by the U. S. Government for the lubrication of such parts of motor equipment and other machinery as are lubricated by means of compression cups:

"The grease shall be a well-manufactured product, composed of a calcium soap made from high-grade animal or vegetable oils or fatty acids, and a highly refined mineral oil.

The mineral oil used in reducing the soaps shall be a straight well-refined mineral oil with a Saybolt viscosity at 100 deg. fahr. of not less than 100 seconds.

PROPERTIES AND TESTS

Soap content. — The content of soap for the several grades shall be as follows:

(*a*) No. ½ cup grease shall contain approximately 13 per cent of calcium soap.

(*b*) No. 1 cup grease shall contain approximately 14 per cent of calcium soap.

(*c*) No. 3 cup grease shall contain approximately 18 per cent of calcium soap.
(*d*) No. 5 cup grease shall contain approximately 24 per cent of calcium soap.

Consistence. — These greases shall be similar in consistence to the approved trade standards for No. ½, No. 1, No 3, and No. 5 grease.

Moisture. — The grease shall be a boiled grease, containing not less than 1 nor more than 3 per cent of water when finished

Corrosion. —A clean copper plate shall not be discolored when submerged in the grease for 24 hours at room temperature.

Ash. — The ash content shall be as follows:

No. ½ grease: The ash shall not be greater than 1 7 per cent.
No. 1 grease: The ash shall not be greater than 1.8 per cent.
No. 3 grease: The ash shall not be greater than 2.3 per cent
No. 5 grease. The ash shall not be greater than 3.5 per cent.

Fillers. — The grease shall contain no fillers such as resin, resinous oils, soapstone, wax, talc, powdered mica or graphite, sulphur, clay, asbestos, or any other filler.

All tests shall be made according to the methods for testing lubricants adopted by the Committee on Standardization of Petroleum Specifications."

Government Specification for Greases: Tech. Paper 323, Bureau of Mines, 1922.
Commercial Lubricating Greases Prac Engineer, U. S., Apr., 1911, p. 293; *Tests of Grease Lubrication:* Ibid., p. 295.

323. Solid Lubricants. — Dry graphite, soapstone, and mica are sometimes used as lubricants, though they are usually mixed with grease or oils. They cannot easily be squeezed or scraped from between the surfaces, and are consequently suitable where very great weights have to be carried on small areas and when the speed of rubbing is not high. The coefficient of friction of such lubricants is high, and, when economy of

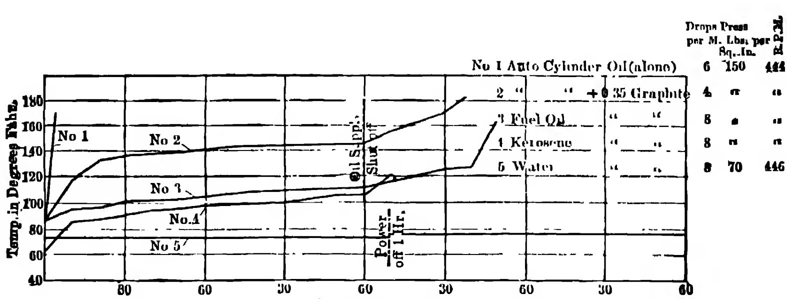


Fig. 595. Tests of Graphite Mixed with Various Lubricants.

power is essential, better results may be secured by the use of liberally proportioned rubbing surfaces and liquid lubricants. Under certain conditions of pressure and speed, these lubricants will sustain, without injury to the surfaces, pressures under which no liquid would work.

Deflocculated graphite suspended in oil or water, and designated commercially as " oildag " and " aquadag " respectively, is finding favor with

many engineers. Graphite in this deflocculated condition remains suspended indefinitely in water and oil, readily adheres to the journal, has great wearing properties, and is easily applied to the wearing surfaces. From numerous and long-continued trials it appears that 0.35 per cent serves adequately for all purposes. Temperature curves of deflocculated graphite in combination with various carrying fluids are given in Fig. 595.

Lubrication with Colloidal Graphite: by C. F. Mabery, Jour. Indus. and Engrg. Chemistry, Vol. 5, No. 9, Sept., 1913.

324. Qualifications of Good Lubricants. — A good lubricant should possess the following qualities:

(1) Sufficient "body" to prevent the surfaces from coming into contact under conditions of maximum pressure.

(2) Capacity for absorbing and carrying away heat.

(3) Low coefficient of friction.

(4) Maximum fluidity consistent with the "body" required.

(5) Freedom from any tendency to oxidize or gum.

(6) A high "flash point" or temperature of vaporization and a low congealing or "freezing point."

(7) Freedom from corrosive acids of either metallic or animal origin.

325. Testing Lubricating Oils. — There is no question but that the lubricant best suited for a given set of conditions can only be determined by an actual practical test under service conditions. Each plant is an individual problem, since certain grades and qualities of oil which work perfectly in some cases have proved entirely unsatisfactory in others where the conditions appeared to be exactly the same. Nevertheless, in order to avoid needless experiment and to limit the number of acceptable lubricants to a minimum, it is desirable to know certain characteristics which will indicate whether or not the particular lubricant under consideration is unfitted for the desired service. The small consumer must depend upon the reputation of the concern from which he is buying for reliable data pertaining to the qualifications of their products, since the cost of conducting a series of preliminary or identification tests is out of all proportion to the actual cost of the lubricant. The large consumer, on the other hand, may find it to be worth while to conduct an elaborate series of tests before drawing up contracts for the oil supply.

All tests should be conducted in accordance with accepted standards, but unfortunately there is no single standard. For general purposes preference should be given to the standards advocated by the American Society of Testing Materials issued triennially by the Society. U. S.

Government specifications must necessarily be followed for lubricants intended for all agencies of the Government.

Report of Committee on Standardization of Petroleum Specifications: Bureau of Mines, Bul. No. 5, 1921.

Specifications for Petroleum Products: Bureau of Mines, Tech. Paper 323, 1922.

The complete test of an oil consists of three parts: Chemical, physical, and practical.

326. Chemical Tests of Lubricating Oils. — In a general sense the great majority of specifications require that all oils should be neutral in reaction and should not show the presence of moisture, matter insoluble in petroleum ether (hard asphalt), matter insoluble in ether alcohol (soft asphalt), free sulphur, charring or wax-like constituents, naphthenic acids, sulphonated oils, soap, resin or tarry constituents, the presence of which indicates adulteration, or lack of proper refining. Except in compound lubricants no traces of fixed oils (animal or vegetable fats) should be found.

Approved fixed oils, such as rapeseed, olive, tallow, lard, and neat's-foot oil, may be used with lubricating oil for main engines without forced lubrication. When the foregoing fixed oils are used, they must be well refined with alkalies, unadulterated, containing a minimum of free fatty acids, with no moisture or gumming constituents. Olive oil should not have a high specific gravity. If satisfactory emulsifying results can be obtained with straight mineral oils on engines without forced lubrication, they may be submitted for service test.

The most satisfactory procedure is to have the various tests made by a competent chemist; but since a number of plants are provided with the necessary equipment, the tests which are conducted by a large central station (and which are representative of current commercial practice) will be described in a general way.

Sulphur. — Boil about 50 cc. of oil with a piece of bright metallic sodium for half an hour; add water, heat and stir until the sodium is dissolved; pour off the water and test the remainder with a fresh 1 per cent solution of sodium nitroprusside. If the mixture turns violet color, the oil contains sulphur. When sulphur is found, the following test for sulphonated oils is made.

Sulphur Test

Approximately 1 gram of oil is weighed into the calorimeter cup and placed in the bomb, which contains 20 cc. of distilled water. The purpose of the water is to absorb the sulphur trioxide formed from the oxidation of the sulphur, converting it to sulphuric acid. The ignition wire is attached to the terminals of the bomb, the center of the wire dipping into the oil. Fine platinum wire should be used for this purpose. The

bomb is then closed and oxygen introduced up to a pressure of 400 lb. per sq. in. When this pressure is obtained the valve of the bomb is closed and the charge is ignited. From 10 to 15 minutes is allowed to elapse for the complete combustion of the oil. The gas in the bomb, after complete combustion has taken place, is allowed to escape slowly.

When the gas in the bomb has been reduced to atmospheric pressure, the bomb is opened and the inside rinsed with distilled water, the washings being collected in a beaker. The solution is then made alkaline, either with ammonium hydroxide or sodium carbonate solution, and heated to completely precipitate any heavy metals, and then is filtered. The filtrate is then acidified with hydrochloric acid, heated to boiling, and barium chloride is added drop by drop until an excess of the precipitant is present. The solution is then allowed to stand for two hours on the hot plate to obtain complete precipitation of the barium sulphate. The precipitate is then filtered, washed, dried, ignited, and weighed as barium sulphate.

Acidity. — (A.S.T.M. D47–18.) Accurately weigh 10 g. of the oil into a flask, add 50 cc. of 95 per cent alcohol which has been neutralized with weak caustic soda, and heat to the boiling point. Agitate the flask thoroughly in order to dissolve the free fatty acids as completely as possible. Titrate while hot with aqueous tenth-normal alkali, free from carbonate, using phenolphthalein, alkali blue, or turmeric as an indicator, agitating thoroughly after each addition of alkali.

To express results as percentage of oleic acid, use the following equation: 1 cc. of tenth-normal alkali = 0.0282 gram of oleic acid. Alkali, 1 cc. of which is equivalent to 0.5 per cent of oleic acid, may be used.

Saponification. — (A.S.T.M. D94–21T.) Weigh 10 g. of oil into a 350-cc. Erlenmeyer flask. Add from a pipette 50 cc. of the alcoholic potassium hydroxide solution followed by 25 cc. of the purified benzene (C_6H_6). Connect with a condenser loop. Boil on steam bath or electric hot plate for 90 minutes, shaking occasionally. Remove and add 25 cc. of neutral gasoline, and titrate with the half-normal hydrochloric acid solution after adding 2 or 3 drops of the phenolphthalein indicator solution until the pink color is destroyed. The absence of the pink color may be determined after the titration has begun by allowing the solution to stand at rest approximately a minute and noting the color of the lower zone. Run two blanks with the same mixture of alcoholic potassium hydroxide solution and purified benzene. From the difference between the number of cubic centimeters of half-normal acid required for the blanks and for the determination, the percentage of fatty oil may be calculated as follows:

$$\frac{\text{No. of cc. N/2 acid used} \times .02805 \times 100}{.195 \times \text{weight of oil taken}} = \text{per cent of fatty oil.}$$

To Detect Fixed Oils. — Heat 10 cc. of oil with a small piece of metallic sodium. If the mixture becomes gelatinized or a semi-solid, it indicates the presence of fixed oils. If an equal volume of oil is heated alone to the same temperature, the viscosity of the two samples can be compared; if the oil contains fixed oils (animal or vegetable oils), the sample with sodium will be much heavier than the sample heated alone.

Effect of Heat. — Heat 5 cc. of oil in test tube over flame until vapors are evolved, and compare the color of the heated oil with that of unheated oils. If the heated oil turns black, it shows the presence of undesirable carbon or hydrocarbons.

Carbon Residue. — (A.S.T.M. D47–21.) The tests shall be conducted as follows: Ten grams of the oil to be tested are weighed in the porcelain crucible *a* which is placed in the Skidmore crucible *b*, and these two crucibles set in the larger iron crucible *c*, care being taken to have the Skidmore crucible set in the center of the iron crucible, covers being applied to the Skidmore and iron crucibles. Place on triangle and suitable stand with asbestos block, and cover with sheet-iron or asbestos hood in order to distribute the heat uniformly during the process.

Heat from a Bunsen burner or other burner is applied with a high flame surrounding the large crucible *c* until vapors from the oil start to ignite over the crucible, when the heat is slowed down so that the vapor (flame) will come off at a uniform rate. The flame from the ignited vapors should not extend over 2 in. above the sheet-iron hood. After the vapor ceases to come off, the heat is increased as at the start and kept so for five minutes, making the lower part of the large crucible red hot, after which the apparatus is allowed to cool somewhat before the crucible is uncovered. The porcelain crucible is removed, cooled in a desiccator, and weighed.

The entire process should require one-half hour to complete when heat is properly regulated. The time will depend somewhat upon the kind of oil tested, as a very thin, rather low flash-point oil will not take as long as a heavy, thick, high flash-point oil.

Corrosion Test. — A clean strip of pure copper about 1/2 in. wide and 2 in. long is heated to redness in a Bunsen flame, and while red hot is dropped into alcohol. The strip is then allowed to dry as quickly as possible in the air and dropped into a sample of the oil contained in a test tube. About half the length of the copper strip should be submerged. The test tube is then closed with a stopper and left to stand for twenty-four hours. At the end of this time, the copper strip is removed and washed clean with proper solvents. It is then compared with a similar strip freshly cleaned as previously described. No discoloration of the test strip should be shown by this comparison. See A.S.T.M.,1921, p. 701.

Reaction Test. — Place 50 cc. of the sample and 15 cc. of distilled water in a 150 cc. flask. Warm to 150 deg. fahr. and shake thoroughly. Allow the mixture to cool and transfer 5 cc. of the aqueous layer to each of two test tubes, by means of a pipette. Add 1 drop of 1 per cent solution of methyl orange to the contents of one tube, and 1 drop of 1 per cent solution of phenolphthalein to the other. No red or pink color should result in either case.

327. Physical Tests of Lubricating Oils. — The physical characteristics usually involve (1) color; (2) odor; (3) specific gravity; (4) flash point; (5) fire point; (6) cold point; (7) viscosity; (8) emulsion; (9) evaporation; and (10) friction. The following tests, unless otherwise indicated, refer specifically to the requirements of the Navy Department which, as previously stated, are representative of current commercial practice.

Color. — The color, although having no influence on the lubricating value, may be used to identify the sample. American oils fluoresce with a grass-green color; Russian oils have a blue sheen; oils containing distillation residues and unfiltered oils are brown to green-black in reflected light. Nearly all mineral machinery oils are distilled and filtered to some extent and are transparent in a test tube, the colors ranging from a yellowish white to a blood red. The color may be determined in a tinctometer by comparing with different-colored glasses or lenses. These glasses are numbered, and for machinery oil extend from No. 1 (white) to No. 6 (red). Consult Tech. Paper 323, Bureau of Mines, 1922, p. 31.

Odor. — The odor may be determined by heating in a test tube or by rubbing on the hand, by which means fatty oils, coal tar, rosin oils, etc., may be detected.

Specific Gravity. — The specific gravity may be determined by the use of the Westphal balance, hydrometer, or "pyknometer," this term signifying any vessel in which an accurately measured volume of liquid can be weighed. When using the pyknometer, the bottle is first filled with distilled water at a temperature of 60 deg. fahr., and the weight of the water determined. The bottle is then filled with oil at a temperature of 60 deg. fahr. and the weight of the oil determined. The weight of the oil divided by the weight of the water gives the specific gravity at 60 deg. fahr. The Baumé gravity is obtained by using the Baumé hydrometer, which is simply an ordinary hydrometer with a certain arbitrary scale. Baumé gravity may be converted into specific gravity by the following formula:

$$\text{Sp.gr.} = \frac{140}{130 + \text{Baumé}} \tag{281}$$

Baumé gravity is largely used in commercial practice.

The specific gravity does not affect the lubricating value of an oil, but it indicates to the experienced oil man the locality from which the crude oil is obtained. For instance, a Baumé gravity of 32 corresponds to a specific gravity of 0.864, and a Baumé gravity of 18.1 to a specific gravity of 0.945, so that an increase in specific gravity is a decrease in Baumé gravity. The paraffin-base oils of Pennsylvania derivation have an average specific gravity of 0.875 with a corresponding Baumé gravity of 30. The asphaltic-base oils from Texas and California have an average specific gravity of 0.930 with a corresponding Baumé gravity of 20.

Flash Point. — (A.S.T.M. D93–22.) The flash point is determined with both the Cleveland open cup and the Pensky-Martin closed cup. The flash point of all oils is determined as a measure of their volatility. The flash point of steam-cylinder oils is of primary importance, the required flash point depending on the temperature of the steam at the engine. With lubricating oils for bearings, the flash point is important only in that it indicates the volatility of the oils and the presence of kerosene or naphtha fractions, with the accompanying fire risks. In the case of very low flash-point lubricating oils, it is desirable to run a special distillation or volatility test, mentioned under chemical tests. The flash point determined with the open cup is higher than with the closed cup, as the inflammable gases on the surface of the oil are disturbed by the air currents in the open cup. These differences range from 5 deg. to 40 deg., with the average at 20 deg. The presence of very light ends (kerosene, naphtha, etc.) may increase this difference to 100 deg.

TABLE 103

SPECIFIC GRAVITY AND BAUMÉ GRAVITY OF A NUMBER OF LUBRICANTS

	Specific Gravity	Gravity Baumé	Flash Test Deg. Fahr.
Water	1.000	10	....
Cylinder oil	9090	24 5	575
Cylinder oil	8974	26	540
Heavy engine oil	9032	25 5	411
Medium engine oil	9090	24	382
Light engine oil	.8917	27	342
Castor machine oil	8919	27	324
Lard oil	9175	23	505
Sperm oil	.8815	29	478
Tallow oil	9080	24.5	540
Cottonseed oil	9210	22	518
Linseed oil	.9299	19	505
Castor oil (pure)	9639	15	....
Palm oil	9046	25	405
Rape-seed oil	9155	23	....
Spindle oil	.8588	33	312

Fire Point. — This is the temperature at which the oil burns, and is determined by raising the temperature about 3 deg. a min., applying the flame for about a second. The fire, or burning, point is from 30 deg. to 65 deg. higher than the flash point with all lubricating oils, the light oils having a difference of about 40 deg.

Cloud and Pour. — Mineral oils become more viscous on cooling, and finally solidify. In lubricating oils refined from paraffin-base crudes, cooling first causes the paraffin particles to solidify, which gives the oil a cloudy appearance; with this class of oils this change is known as the cloud point. The A.S.T.M. instructions are as follows: Take a bottle about 1 1/4 in. inside diameter and 4 to 5 in. high, and pour in oil to a height of 1 1/4 in. from the bottom. Insert a cold-test thermometer (especially made, using colored alcohol, and with a long bulb) through a tight-fitting cork. A special jacket is used, having an inside diameter about 1/2 in. larger than the bottle. Ice or any other cooling medium is packed around this jacket. When the oil is near the expected cloud point, at every 2 deg. drop in temperature remove the bottle and inspect the oil, being careful not to disturb the oil. When the lower half becomes opaque, read the thermometer; this reading is taken as the cloud point. The cold, or pour, test is simply a continuation of the cloud test, except that the temperature is noted every 5 deg. and the bottle tilted till the oil flows. When the oil becomes solid and will not flow, the previous 5-deg. point is taken as the cold point of the oil.

Cold Point. — The object of this test is to determine the lowest temperature at which oil will flow from one end of a container to the other. In case it should become frozen the resulting solid oil is stirred till it has assumed a sufficiently pasty consistency to flow. The test is conducted by freezing an ounce of the oil solid in an ordinary 4-oz. oil-sample bottle, using a freezing mixture if necessary. The frozen oil is thoroughly stirred with a thermometer until the mass will run from one end of the bottle to the other, and at this moment the temperature as indicated is recorded.

Emulsion Tests. — The oil and water to be emulsified are contained in an ordinary commercial 100-cc. graduated cylinder, 1 1/16 to 1 2/16 in. inside diameter. An oil or water bath is provided for maintaining the contents of the cylinder at a temperature of 130 deg. fahr., except when a different temperature is specified, both during the stirring and during the subsequent settling out of the oil from the emulsion. The paddle used in stirring is a copper plate 4 3/4 in. long, between 3/4 and 7/8 in. wide and 1/16 in. thick. Means are provided for revolving this paddle about a vertical axis parallel to and midway between its two longer edges and for keeping the speed fairly constant at 1500 r.p.m.

Some form of holder for the cylinders is a convenience but not a necessity, since on account of the ample clearance between paddle and cylinder and the fact that a sample is stirred for only five minutes, a cylinder may be held by hand during the stirring. A stop should be provided so that when the paddle is lowered into the cylinder (or bath raised) the distance from the bottom of the paddle to the bottom of the cylinder will be about 1/4 in. To save time that would otherwise be lost in waiting for the filled cylinders to come to the temperature of the bath, it is desirable that the bath should be large enough to contain several cylinders.

Forty cc. of the emulsifying liquid is placed in a clean 100-cc. graduated cylinder, and 40 cc. of the oil to be tested is added. The cylinder is then placed in the bath, and when the contents have reached the temperature required for the test they are stirred by the paddle for five minutes. The paddle is stopped, withdrawn from the cylinder, and wiped clean. The cylinder is then allowed to stand for the specified time and is then inspected.

Emulsification of Mineral Lubricating Oils Power, Oct. 29, 1918, p. 649.

Demulsibility Test. — Pour 27 cc. of the oil to be tested and 53 cc. of distilled water into a cylinder, place cylinder in bath and heat to 130 deg. fahr. Submerge the paddle and run it for five minutes at a speed of 1500 r.p.m. Stop the paddle, withdraw it from the cylinder, and use the finger to wipe off the emulsion clinging to the paddle and to return it to the cylinder. Wipe off the paddle with paper so that it will not contaminate the next sample. Keep the temperature of the cylinder constant at 130 deg. fahr. and take readings every minute of the position of the line of demarcation between the topmost layer of oil and the adjoining emulsion. The first reading is taken one minute after stopping the paddle. With oils which act normally, the rate of settling out of the oil increases up to a maximum and then decreases, and the maximum value in cc. per hour is called the "demulsibility" and is recorded as the numerical result of the test. Each rate of settling is the average rate calculated from the time of stopping the paddle to the time of reading, as shown in the following condensed table:

Time	Time since Stopping Paddle, Mins.	Reading at Interface Between Oil and Emulsion	Oil Settled Out, cc	Rate of Settling, cc per Hr.
9.50	0	80	0	0
9.55	5	77	3	36
10 02	12	67	13	65
10.05	15	63	17	68
10.10	20	61	19	57

The demulsibility in this case would be 68, the highest value in the last column. In cases where the maximum rate of settling has not been reached at the end of one hour, the test is discontinued and the demulsibility taken as the number of cc. that settled out in the hour.

Precipitation Test. — Five cc. of the oil is mixed with 95 cc. of petroleum ether in a tall, stoppered, graduated cylinder, and allowed to stand. The petroleum ether must be freshly redistilled and the portion boiling above 150 deg. fahr. discarded. It must not show perceptible solubility in concentrated sulphuric acid.

Viscosity. — Experimental evidence indicates that under conditions experienced between flat lubricated surfaces the oil flows in parallel layers much like a pack of playing cards sliding over each other, the outer layers adhering to the surfaces and not sliding with respect to them. The resistance of these layers to sliding past each other is due primarily to fluid friction or so-called **absolute viscosity** and is defined as the force in dynes necessary to move each sq. cm. of metal surface at a velocity of 1 cm. per sec., if the distance between the surfaces is 1 cm. According to the generally accepted theory of lubrication, when a perfect oil film exists between moving surfaces, so that they do not touch, the bearing friction is due solely to the absolute viscosity of the lubricant. That this is not strictly true is evidenced by the experiments conducted by Dr. Stanton, in which the addition of 1 per cent of oleic acid to Bayonne oil, an amount insufficient to produce a noticeable effect on the absolute viscosity, reduced the coefficient of friction in a bearing by no less than 17 per cent. Absolute viscosity, nevertheless, is one of the more important characteristics of a lubricant, but unfortunately it is difficult to measure directly. The time, however, which is required for a given mass of liquid to gravitate through an orifice is a function of the absolute viscosity, so that it is only necessary to determine this time factor for comparison of relative absolute viscosities. The **Saybolt Universal viscosimeter** conforming to the dimensions specified by the U. S. Bureau of Standards is the recognized standard in this country. (See A.S.T.M. Standards, 1920, p. 703, for detailed description.) Saybolt viscosities are expressed as the number of seconds required for 60 cc. of fluid to pass through the orifice at a specified temperature. The temperatures usually employed are 100 and 130 deg. fahr. for non-viscous oils and 210 deg. fahr. for thick or viscous oils. Since the only force available for driving the fluid through the tube is gravity, and the force holding the lubricant back in the cup is its internal resistance or viscosity, it is apparent that the time required for a given quantity to flow through the orifice will be dependent upon both the absolute viscosity and the density of the fluid. The Saybolt viscosimeter, therefore measures a unit equivalent to the

absolute viscosity divided by density, which is called the **kinematic viscosity,** and which, for the standard dimensions specified by the U. S. Bureau of Standards,[1] may be expressed

$$K = v/d = 0.0022\,t - 180/t \tag{282}$$

TABLE 104

PHYSICAL CHARACTERISTICS OF A NUMBER OF LUBRICANTS

Kind of Oil	Application	Gravity	Flash	Fire	Pour	Viscosity at		
						100	210	Compound
Superheat valve oil	For steam cylinders using superheated steam at pressures above 150 lb.	22½–25½	525 590	585 650	30 60		150 190	0 to 5%
High-pressure cylinder oil	For steam cylinders using saturated steam pressures 140 to 175 lb.	22½–26	510 575	570 615	30 60		140 160	3 to 6%
General cylinder oil	For steam cylinders using medium pressures saturated steam 100 to 135 lb.	23–26½	470 540	545 600	30 50		120 140	4 to 8%
Low-pressure cylinder oil	For steam cylinders under pressures below 110 lb.; steam usually wet	23–27	460 540	540 600	30 50		95 125	2 to 10%
Engine oil	For external use on engines in drip cups or circulating systems	19½–28	315 410	350 460	0- 30	150 250		
Ice-machine oil	For ammonia cylinders of refrigerating machines	19–30	300 370	340 425	20 0	100 150		
Gas-engine oil	For heavy gas engine and so-called semi-Diesel oil engines	20–27½	360 450	400 500	5 35	450 1000	50 75	
Diesel-engine oil	For cylinders of Diesel engines	20–27	360 500	400 540	5 35	700 2000	55 120	
Heavy journal oil	For heavy, slow-moving parts on dredging and like machinery	18–25	320 440	360 490	0–40	500 2000	50 100	
Automobile-engine oils	Cylinders of automobiles	19–30	300 450	335 525	0–45	140 2000	40 120	Range of all grades L to XX heavy
Marine-engine oils	External parts of marine engines	20–28	325 420	370 500	10–45	400 900	50 80	5–15% Blown Rapeseed
Cutting oil	Cutting and cooling	19–28	315 420	350 470	0–30	125 300		5% to 4% Lard Oil

[1] Tech. Paper, No. 112, 1919.

in which

K = kinematic viscosity, centipoises (the viscosity of water at 68.4 deg. fahr. is 1 centipoise),
t = Saybolt time, in seconds,
V = absolute viscosity,
d = specific gravity.

Absolute viscosities are used in making any calculation on frictional resistances, and Saybolt viscosities in comparing the physical properties of lubricants.

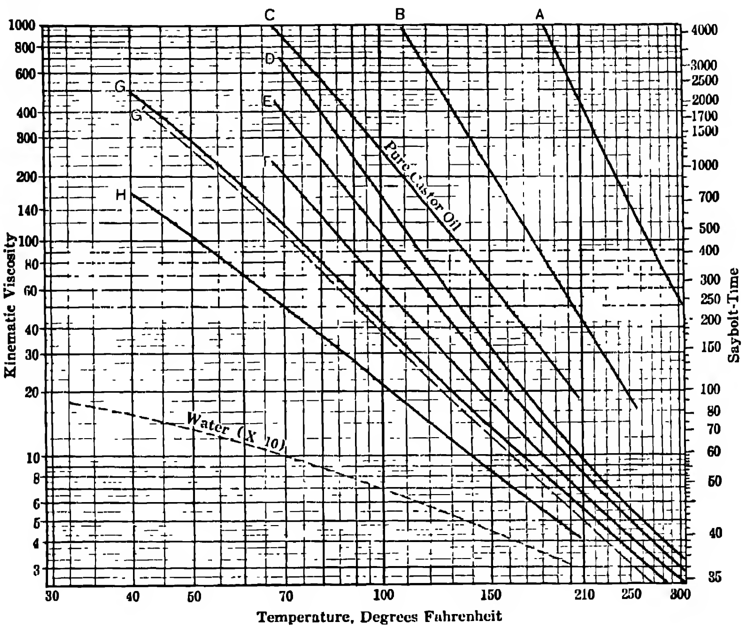


Fig. 596. Influence of Time on Viscosities of a Number of Lubricants.

The influence of temperature on the kinematic viscosity and the corresponding Saybolt time, for a number of lubricants, is shown in Fig. 596.

Other makes of viscosimeters which are frequently used in determining viscosities are the **Redwood** of Great Britain, the **Engler** of Germany, and the **Barbey** of France. For comparisons of the readings of the various types of viscosimeters, see Bureau of Standards, Tech. Paper 112, 1919, p. 21. The curves in Fig. 597 are of interest in showing the relation between absolute viscosities and viscosities as determined from the Saybolt, Redwood, Engler, and Barbey viscosimeters. It will be seen from the curves that the viscosity varies considerably with the temperature;

therefore, the engineer should determine the operating temperature as accurately as possible, and then select the lubricant that will have the correct viscosity at that temperature.

Standardization of the Saybolt Universal Viscosimeter: Bureau of Standards, Tech. Paper No. 112, 1919.

The Saybolt Viscosimeter: Power, March 7, 1922, p. 376.

How Variation of Temperature Affects Viscosity of Lubricating Oils: Power, March 14, 1922, p. 420.

Viscosity: Lubrication (Texas Co.), Vol. 6, No. 6, July, 1920.

Kinematic Viscosity = Absolute Viscosity ÷ Specific Gravity

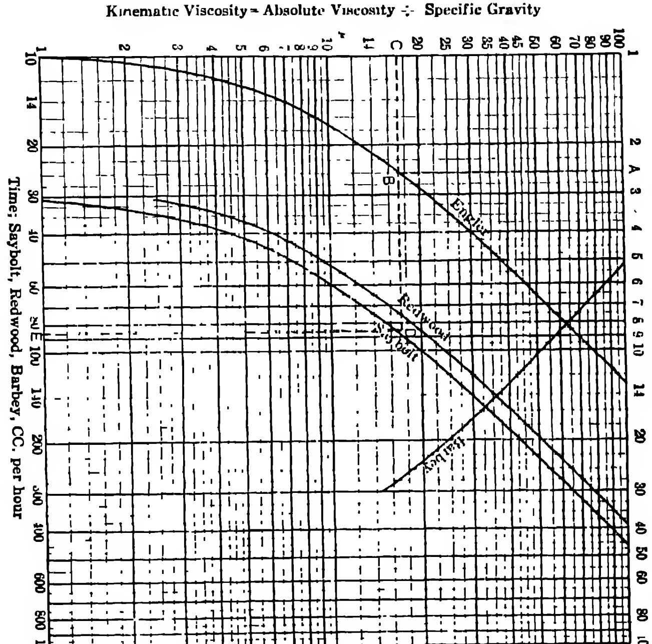


FIG. 597. Relation between Absolute and Indicated Viscosity.

Friction Tests. — The coefficient of friction, as determined from friction-testing machines, is useful in obtaining a comparison of oils under the test conditions, but gives little information concerning the action of the oil under the widely different conditions found in actual practice.

Table 104 gives the physical properties of a number of lubricating oils, with their particular fields of application.

Service Tests. — These tests are the real proof of the commercial value of the lubricant for a given service. The lubricants are tested under actual operating conditions, and the one that gives the best overall economy is selected, such factors as first cost, quantity used, effect on the rubbing

surfaces, maintenance and attendance being taken into consideration. After these tests have determined the particular grade of lubricant which gives the best returns, the tests previously mentioned are made, and the results are incorporated in the specifications so as to insure delivery of that particular grade of lubricant. Large consumers frequently employ the services of an experienced lubricating engineer under the supervision of the plant engineer or millwright, for determining the most suitable lubricant for the different classes of machinery.

A Graphical Study of Journal Lubrication: Mech. Engrg., Feb., 1924, p 77.

328. Steam Engine Lubrication — Atmospheric. — In a general sense all journals, slides, and "atmospheric" surfaces should be lubricated with straight mineral oils and greases similar to those specified in Tables 101–103. Bearings, guides, and all external rubbing surfaces requiring oils may be lubricated in a number of ways: (1) they may be given an **intermittent** application of oil, as, for example, with an oil can; (2) they may be equipped with oil cups with **restricted** rates of feed; and (3) they may be **flooded** with oil. The relative lubricating values of the systems have been estimated approximately as follows:

	Coefficient of Friction	Comparative Value
Intermittent	0 01 and greater	72 and less
Restricted feed	0 01 to 0.012	79 to 86
Flooded bearing	0 00109	100

Intermittent Feed. — Intermittent applications are ordinarily limited to small journals, pins, and guides which are subject to light pressures and which do not easily permit of oil or grease cups, as, for example, parts of the valve gear of a Corliss engine, governors, and link work. On account of the labor attached and the frequent doubt about the oil reaching the wearing surfaces, this method of lubrication is limited as much as possible even in the smallest plants.

Restricted Feed. — In the average power plant the major part of the lubrication is effected by means of oil cups which are filled at intervals by hand or by mechanical means, the oil being fed from the cup by drops, according to the requirements.

Oil Bath. — In large power plants the principal journals and wearing parts are supplied with a continuous flow of oil which completely "floods" the rubbing surfaces. The oil is forced to the various parts either by gravity from an elevated tank or by pressure from a pump. After the oil leaves the bearings, it flows into collecting pans, thence into a receiving and filtering tank, and finally is pumped back into an elevated reservoir

and used over and over again. The little lost by leakage and depreciation is replenished by the addition of new oil to the system.

Oil Cups. — Figure 598 illustrates the application of **sight-feed** oil cups to the crosshead and slides of a reciprocating engine. The oil is fed into the cups by hand and gravitates to the rubbing surfaces, the rate of flow being regulated by a needle valve. Cups *A* and *B* feed directly to the crosshead guides, but the oil from cup *D* flows to the bottom orifice *O*, from which it is wiped by a metallic wick *S*, and carried by gravity to the wrist pin.

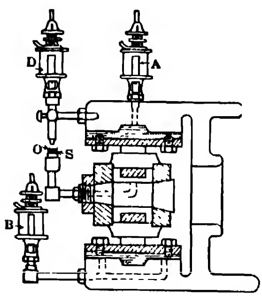
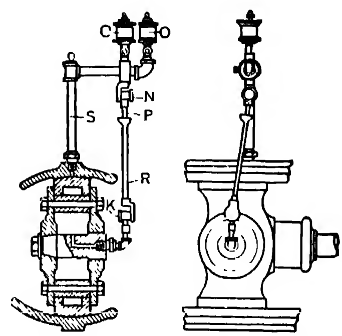


FIG. 598. Oil-cup Lubricator, Hand-filled.



599. Typical Telescopic Oiler.

Telescope Oiler. — Figure 599 shows the application of a **telescopic oiler** to a crosshead and guides. *O* and *C* are sight-feed oil cups, the former feeding directly to the top guide through the tube *S*. The oil from *C* flows by gravity through the swing joint into the telescopic tubes *P*, *R*, and thence to the pin through the lower swing joint as indicated. As the crosshead moves back and forth, the pipe *P* slides into and out of pipe *R*, the oil being thus conducted directly to the pin without wasting. A device of this type installed on a high-speed automatic engine at the Armour Institute of Technology has been in operation for five years without cost for repair or renewal.

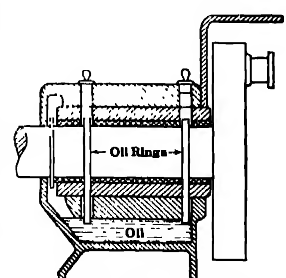


FIG. 600. Oil-ring Lubrication.

Ring Oiler. — Small high-speed engines are often oiled by the **oil-ring** system, as illustrated in Fig. 600. The shaft is encircled by several loose rings which dip into a bath of oil in the case of the pedestal or frame and, rolling on the shaft as it turns, carry oil to the top of the shaft where it spreads to the bearings. In some cases the rings are replaced by loops of chain.

Ring Lubrication: Power, May 2, 1922, p. 687; Jan. 9, 1917, p. 42.

TABLE 105

U. S. GOVERNMENT SPECIFICATIONS FOR LUBRICATING OILS

(1923)

Name and Grade	Flash, Minimum	Fire, Minimum	Viscosity, Saybolt Seconds 100 Deg. Fahr.		Color, NPA, Darkest Color Allowed When Diluted 50 Per Cent	Pour, Maximum	Acidity, Maximum Mg KOH per Gram	Corrosion Test	Emulsion Test	Demulsibility, Minimum	Carbon Residue, Maximum	Special tests Required
			Minimum	Maximum								
	°F.	°F.	Sec.	Sec.		°F.					P ct	
Class A, extra light	315	355	135	165	No. 5	35	.....	Required	.	.....	. .	Reaction test
Class A, light	325	365	180	220	"	35	. . .	"	......	... .	. .	
Class A, medium	335	380	270	330	"	40	.	"		. .	.	
Class A, heavy	345	390	360	440	"	45		"	. .	. . .	.	
Class A, extra heavy	355	400	450	550	No. 6	50		"	.	. . .	. . .	
Class B, extra light	315	355	135	165	No. 5	35		"	Required	300	.	
Class B, light	325	365	180	220	"	35		"	"	300	.	
Class B, medium	335	380	270	330	"	40		"	"	300	. . .	
Class B, heavy	345	390	360	440	"	45	. .	"	"	300	. . . .	
Class B, extra heavy	355	400	450	550	No. 6	50		"	"	300		
Class C, extra light	315	355	135	165	No. 5	35	0 10	"	"	300	0.10	
Class C, light	325	365	180	220	"	35	10	"	"	300	.20	
Class C, medium	335	380	270	330	"	40	10	"	"	300	.30	
Class C, heavy	345	390	360	440	"	45	10	"	"	300	.40	
Class C, extra heavy	355	400	450	550	No. 6	50	.10	"	"	300	.60	
Class D extra light	315	355	135	165	No. 5	35	.30	"	.	Optional	10	
Class D, light	325	365	180	220	"	35	.30	"	.	"	20	
Class D, medium	335	380	270	330	"	40	30	"	.	"	30	
Class D, heavy	345	390	360	440	"	45	.30	"	. . . .	"	40	
Class D, extra heavy	355	400	450	550	No. 6	50	.30	"	. . . . .	"	.60	

Centrifugal Oiler. — Figure 601 illustrates the application of a **centrifugal oiler** to a side-crank engine. The oil supply is regulated by the sight-feed cup *C* and flows by gravity to the pipe *P* in line with the center of the crank shaft. Centrifugal force throws the oil outward through pipe *B* to the center of the pin *D*, which is drilled longitudinally and radially so as to distribute the oil upon the bearing surface.

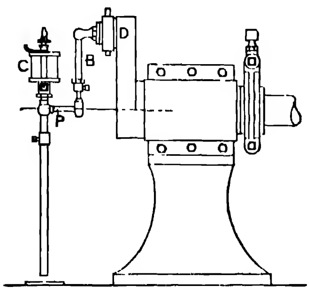


FIG. 601. Centrifugal Oiler.

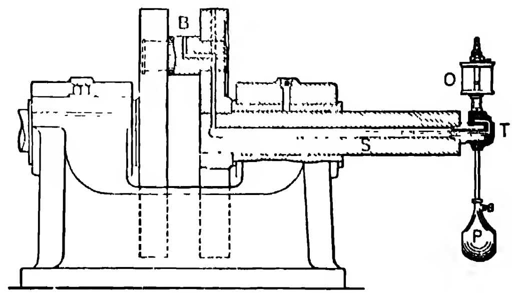


FIG. 602. Pendulum Oiler.

Pendulum Oiler. — Figure 602 illustrates the application of a **pendulum oiler** to the crank pin of a center-crank engine. Oil cups and pendulum *P* are fastened to the crank shaft *S* by trunnion *T*. The pendulum holds the cup vertical, since the friction of the trunnion is not sufficient to revolve it. Oil flows along the center of the crank shaft under the head of oil in cup *O* and is thrown outward to bearing *B* by centrifugal force.

Splash Oiling. — In most high-speed engines, the crank, connecting rod, and crossheads are enclosed by a casing, the bottom of which is filled with oil to such a depth that, at each revolution of the crank, the end of the connecting rod is partly submerged. The result is that the oil is splashed into every part of the chamber, and the crank pin, crosshead pins and crosshead slides practically run in an oil bath.

329. Steam Engine Lubrication — Internal. — All rubbing surfaces which come in contact with the steam must necessarily be lubricated, but it is far more difficult to ascertain whether or not a steam cylinder is efficiently and economically lubricated than to determine the same condition with bearings, because the inside surface of the cylinder cannot be felt while the engine is running. Wear in a cylinder cannot usually be detected except on examination when the cylinder head is removed. As cylinders and valves cannot be examined every day there is a tendency to use an excess of oil.

There are two methods of applying internal lubrication (1) the direct system, and (2) the atomization system. In the former, the lubrication is applied to each of the separate surfaces, while, in the latter, it is injected

into the steam so that the latter acts as a carrier. In either of these systems, the oil may be fed to the desired point by **hydrostatic** or by **mechanically operated** lubricators.

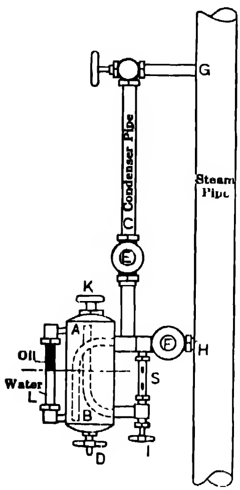


Fig. 603. Common Hydrostatic Lubricator.

Hydrostatic Lubricators. — The earliest method of cylinder lubrication, and one which is still used to a large extent, is that using **hydrostatic lubricators** of the sight-feed class, Fig. 603. The principle of operation is as follows: The lubricator is filled with cylinder oil by removing cap K, the height of oil appearing in glass L. If water is present, the oil floats on top as indicated. After the cap is screwed in place, the valves in the condenser pipe are opened, subjecting the oil in the vessel to steam-pipe pressure. Steam is condensed in pipe C, filling tube B and part of C, thus adding to the steam pressure the pressure due to the weight of the water column. Valve F, which communicates with the top of the vessel by means of tube A, is opened wide, as is also the regulating valve I. The pressure at B, being greater than that at A by an amount equivalent to the height of the water column, forces the oil through A and the "sight-feed" S to the steam pipe. The rate of flow is controlled by the regulating valve I. As the oil flows from the vessel, its space is occupied by condensed steam, the height of oil and water being visible in glass L.

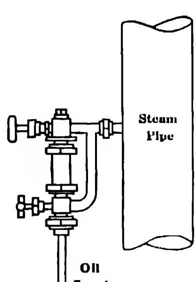


Fig. 604. Sight-feed Lubricator, Central System.

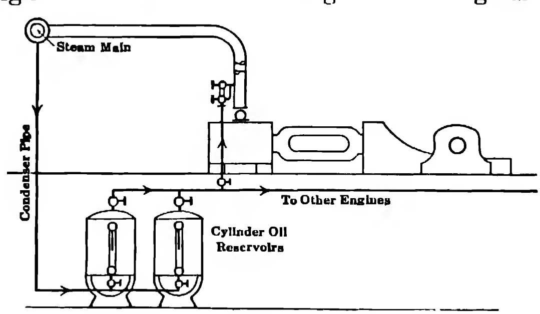


Fig. 605. Hydrostatic Lubrication, Central System.

Owing to the small capacity of the lubricator, it must be refilled frequently. To reduce the amount of labor required with the above apparatus, independent sight-feeds, Fig. 604, are sometimes used in connection with a central reservoir. Such an installation is shown diagrammatically in Fig. 605. A condenser pipe leading from the steam main enters the

bottom of the reservoir, and the condensed steam fills up the reservoir as fast as the oil is fed out. The principle is the same as that of the simple hydrostatic lubricator.

Forced-feed Cylinder Lubrication. — Modern conditions of high-pressure and high-temperature steam make it desirable, and in many cases necessary, to use mechanically operated lubricators which can be relied upon for automatically feeding the lubricant uniformly and in sufficiently small quantities. Figure 606 illustrates the "Rochester" simple feed automatic lubricating pump, which takes the oil by gravity from the reservoir through a sight-feed glass and forces it through a small pipe to the steam supply pipe. The pump entirely obviates the trouble due to intermittent feeding and, being directly driven by the engine, runs at constant speed. The feed is uniform and independent of the pressure pumped against. The rate is determined by the length of stroke of the pump piston which is easily adjusted.

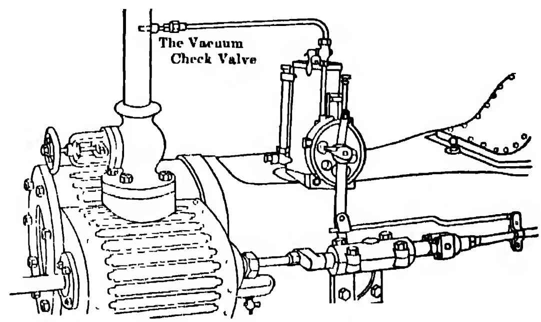


FIG. 606. Forced-feed Lubricator, Independent Type.

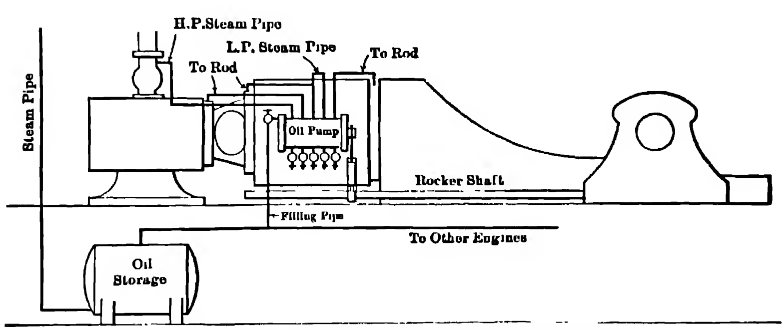


FIG. 607. Forced-feed Lubrication, Central System.

With large engines multi-feed pumps are sometimes used, which force oil to the various valves as well as to the steam pipe. Figure 607 shows an arrangement of storage tanks in connection with pump reservoir to avoid the trouble of hand filling.

See Table 104 for physical characteristics of cylinder oils suitable for internal lubrication.

Steam Cylinder Lubrication: Power Plant Engrg., July 1, 1923.

TABLE 106

U. S. GOVERNMENT SPECIFICATIONS FOR GREASES

(1923)

Name and Grade	Viscosity at 100 Deg Fahr. of Mineral Oil. Minimum	Calcium Soap Content, Approximate	Moisture, Maximum	Corrosion Test	Ash	Soda Soap Content, Minimum	Free Alkali. NaOH	Color	Water, Glycerin, and Impurities, Maximum of Dry Soap Content
	Saybolt Seconds	Per Cent	Per Cent		Per Cent	Per Cent	Per Cent		Per Cent
Cup grease No. 0	100	13	3	Required	1.7	.....	.....	......	....
Cup grease No. 1.....	100	14	3	"	1.8	.....	.....	.....	....
Cup grease No. 3 ...	100	18	3	"	2.3	.....	.....	.....	....
Cup grease No. 5. ...	100	24	3	"	3.5	.....	.....	.....	....
Recuperator grease ..	180	18	3	"	2.3	.....	.....	.....	....
Crank-pin grease ..	.....	.....	.....	.....	.....	40	0.5 to 2.5	Yellowish	33⅓
Driving-journal compound	.....	.....	.....	.....		45	.5 to 2.5	Greenish	33⅓
Rod-cup grease.. .	.....	.....	.....	.		40	.5 to 2.5	Yellowish	33⅓

330. Oil Handling Systems — Steam Engine Plants. — Gravity oil-feed: Fig. 608, illustrates a simple **gravity oil-feed** system. The oil is supplied to the engine from the oil tank, by pipe D under pressure corresponding to the height of the tank above the oil cups. After performing its function the oil gravitates to the filter and from the latter to the oil reservoir, from which it is pumped back to the supply tank, the overflow being returned to the reservoir through pipe N. Operation is interrupted only when new oil is to be added to the system from the barrel through the flexible filling pipe. In case the oil tank is put out of commission, or the supply pipe becomes clogged, full pump pressure may be used by closing valves R and S and opening valve E. The make-up oil is small in amount compared to the quantity circulated. The reclaiming and purifying of the oil are essential if the bearings are to be flooded, otherwise the cost of oil would be prohibitive.

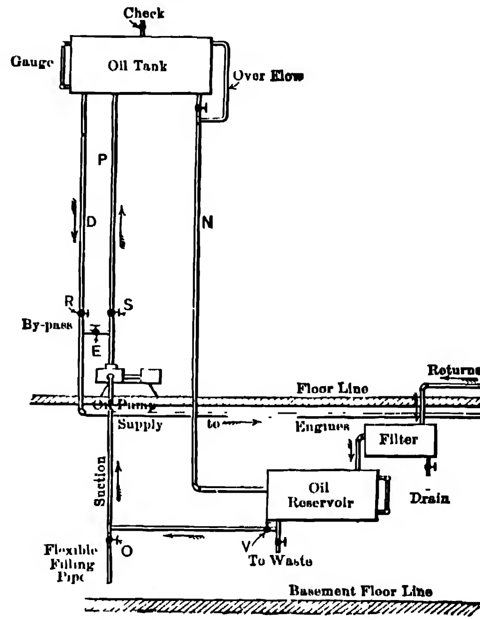


Fig. 608. Simple Gravity-feed System.

An objection sometimes made to the above system is that the varying heights of oil in the supply tank may cause considerable variation in pressure at the oil cups, causing them to feed faster when the tank is full and slower when the tank is nearly empty. This applies only to installations where the supply tank is filled intermittently.

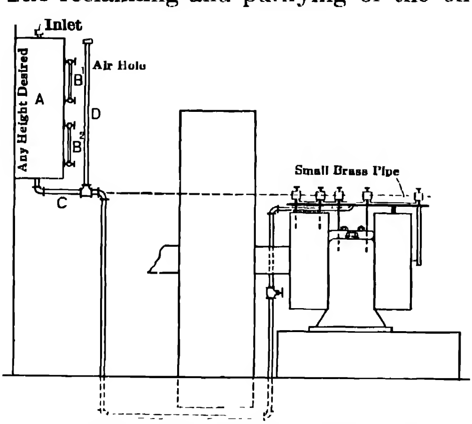


Fig. 609. Low-pressure Gravity Feed, Constant Head.

Figure 609 shows the application of a low-pressure oiling system in which the level in the sight feeds is kept constant. A is the main

supply tank, B^1 and B^2 the upper and lower gages indicating the oil level, C the supply pipe running to the engines, and D a small standpipe closed at one end and vented near the top. The reservoir is supplied with oil by the valve marked "inlet." When the tank is filled, the oil rises in the standpipe D a corresponding height. The inlet valve is then closed and the oil in the standpipe feeds down to the level of the sight feeds or to a point where the air will enter the bottom of the tank. This will be the constant oil level, since oil flows from the tank only in proportion to the amount of air admitted. A head of 6 in. has been found to give the best results.

Compressed-air Feed. — Figure 610 shows diagrammatically the arrangement of an oiling system involving the use of compressed air as the motive power. The storage tank containing the supply of engine oil is under air pressure at all times except during the short periods when it is

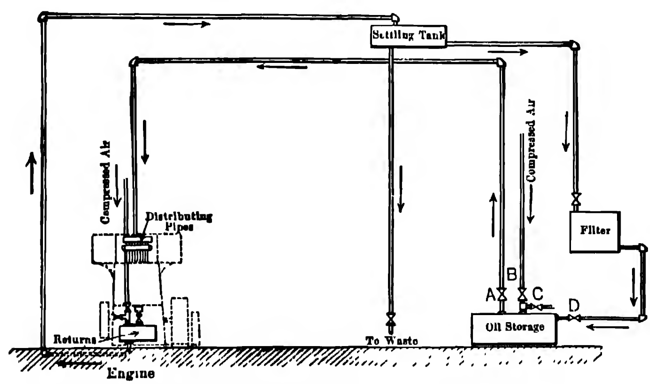


FIG. 610. Compressed-air Feed Oiling System.

being filled with oil from the filter. The air pressure on the surface of the oil forces it to a manifold on the engine from which it is distributed to the various oil cups. The oil flows from the different bearings to the returns tank located at the base of the engines. When the tank is filled, air pressure is admitted and the oil forced to the settling tank, which has a capacity of about 400 gal. and is located near the ceiling. The oil is allowed to settle and the entrained water and foreign material are drained to waste. The oil gravitates from this tank to a series of Turner oil filters. When a new supply of oil is needed, valves A and B are closed and vent valve C opened, cutting off the supply of air and reducing the pressure to atmospheric. Valve D is then opened and oil flows from the filters to the storage tank.

Mechanical Feed. — Figure 611 shows the piping for a large central system of cylinder and engine lubrication, illustrating current practice.

There are two storage tanks on the engine-room floor, one for cylinder oil and the other for engine oil, the distributing arrangements being the same in each case. The oil is pumped from each tank into a main pipe extending the length of the engine room and provided with branches at

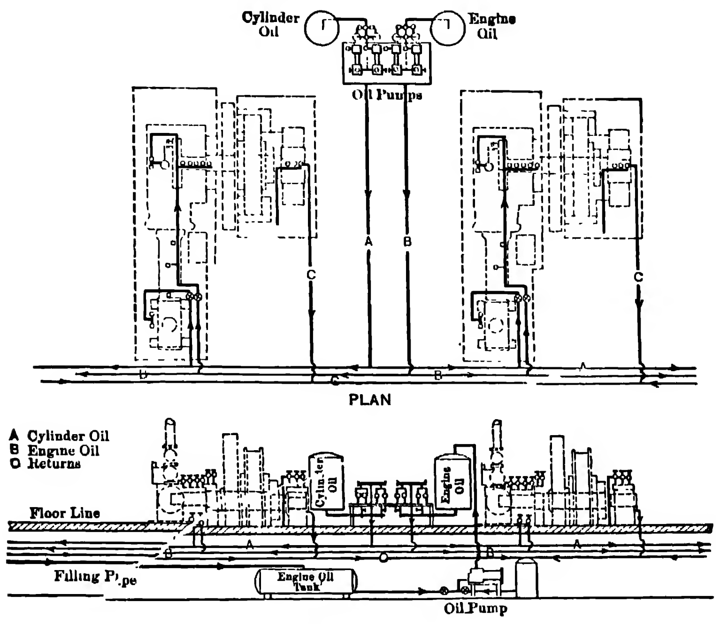


FIG. 611. Central System for Large Stations.

each point requiring lubrication. The oil pumps are actuated by steam and are of the duplex direct-acting type, provided with automatic governors which regulate the speed to suit the demand for oil. The cylinder oil is forced through a special sight-feed lubricator, Fig. 612, under a pressure about 25 lb. in excess of the steam pressure. Referring to Fig. 612, diaphragm valve *D*, in the bottom of the lubricator, is kept closed by the steam pressure admitted through pipes *B*. Thus the inlet pressure must be greater than that of the steam before the valve will open and admit oil to the engine. The oil, after entering, passes upward through the sight-feed glass and downward through the hollow arm *A* to the steam pipe.

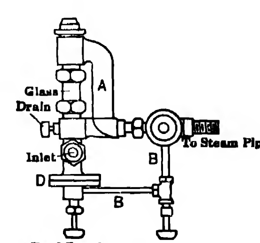


FIG. 612. Sight-feed Lubricator (Forced-feed Type).

The engine oil is forced by the pump to the various points under a pressure of about 20 lb. The waste oil is caught in suitable receptacles and,

after being filtered, is returned to the storage tank by a steam pump. This pump is connected so that it can supply the storage tank either from the filter or with fresh oil from a large oil tank in the basement. By this arrangement all handling of oil in the engine room is done away with.

331. Steam Turbine Lubrication. — The oiling system in small turbines under 200 hp. capacity is usually self-contained and requires no special piping or storage tank. Lubrication of journals is effected by means of oil rings riding free on the shaft. Cored water-cooling chambers are commonly placed below the oil-ring reservoirs, with lapped holes for pipe connections. Circulation of water is necessary where high initial

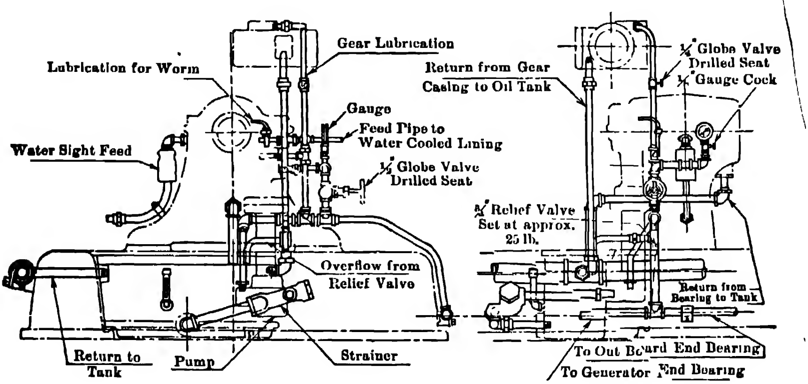


Fig. 613. Diagram of Oil Piping for Small Horizontal Curtis Turbine.

steam temperature conditions prevail and also where a lower oil temperature is advisable or necessary. When the oil loses its lubricating properties, it is removed and the reservoirs are filled with a fresh supply. Small parts, such as the governor levers, ball bearings, and trunnions, are lubricated by hand.

All large turbines and many of the smaller units are supplied with oil from a general oil-lubricating system operating on a continuous-circulation cycle. Each turbine manufacturer has a system peculiar to his product, but in a general sense the cycle is as follows:

Oil is taken from a reservoir located in the bed-plate and forced by a pump (geared to the main shaft or independently driven) through a tubular oil-cooler to the bearings, etc. The warm oil is drained into the reservoir where it is filtered and cooled and from which it is recirculated by the pump. With a mechanically operated governor mechanism, the oil pressure on the system seldom exceeds 25 lb. gage and only one pump is employed. The oil pressures at the points of application vary from 2 to

15 lb. gage depending upon the make and type of turbine. With the oil-relay governor mechanism, an oil pressure of about 50 lb. gage is required for its operation. In some designs there are two pumps, one for the general lubrication and the other for the governor relay. The majority of large turbines, however, have but one pump operating at a pressure high enough for the oil-relay system and furnishing oil for the bearings, etc. at a lower pressure through a reducing valve. All large turbines are equipped with an auxiliary or standby oil pump which may be used when starting and stopping, or in case of emergency.

The " extra light " and " light " turbine oils as specified by the Government (see Table 101) are prescribed by fully 80 per cent of the turbine

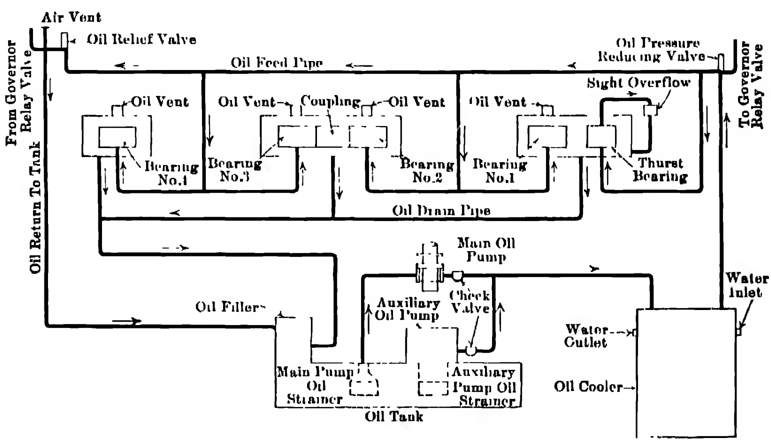


FIG. 614. Lubricating System — Allis-Chalmers Turbine.

plants having forced-feed circulation; however, for ring-oiled turbine sets, the " extra-heavy " oil is used extensively, and in a number of instances, it is necessary to furnish a mineral cylinder oil because of steam leakage. The " heavy " or " extra " turbine oil is commonly specified for reduction gear sets.

Figure 614 gives a diagrammatic outline of the lubrication system of the Allis-Chalmers turbine. The oil reservoir is located in the bed plate, remote from the heated parts of the turbine, and the pump is placed at the high-pressure end. Oil is pumped from the reservoir under a pressure of 25–30 lb. through the cooler and thence to the main distributing pipes. The oil connections for the bearings are located in the lower half of the pedestal, the oil passing through the bottom seat and around the bearing and passing longitudinally along the journals on the horizontal center line and on the top. A sight oil vent is provided above each bearing so as to prevent air from accumulating in the oiling system at the bearings

and also to enable the operating engineer to observe the flow of oil. Part of the oil is used for the relay system of governing the turbine and the remainder, reduced in pressure to about 4 or 5 lb. by a reducing valve, is delivered to the various bearings. Excess pressure in the oil system is controlled by a relief valve which by-passes the oil back to the reservoir.

332. Oil Purification. — Oil used continuously in a lubricating system is subject to deterioration and eventually becomes unfit for further use. The principal cause of this deterioration is oxidation into a product called **sludge,** caused primarily by the action of light, air, water, and heat, though dust, acids and alkalies are also active contributors to its formation.[1] Sludge appears to exist in a soluble and in an insoluble form. The insoluble sludge has a specific gravity greater than that of the oil and will settle at the bottom when sufficient time is allowed, but the soluble compound appears to be in colloidal suspension in the oil at operating temperatures. Most of the oil-purifying devices available on the market will remove practically all of the insoluble sludge, but few will remove the soluble sludge, acids, or alkalies.

There are at present two methods of purifying oil, viz: (1) precipitation and filtration; and (2) centrifugal separation.

Precipitation and Filtration. — Figure 615 shows a section through a small purifier of the **bag type,** suitable for plants where the oil is clarified intermittently. Impure oil is poured into funnel *A* and is discharged below the surface of the water through openings in the foot of the tube. The thin streams of oil rise vertically to the surface of the water, and the heavy particles of grit and dirt gravitate to the bottom. The oil flows from the precipitation chamber through pipe *B* into a filter bag which removes the lighter impurities carried over with the oil. This type of filter does not remove the soluble sludge or the acidity.

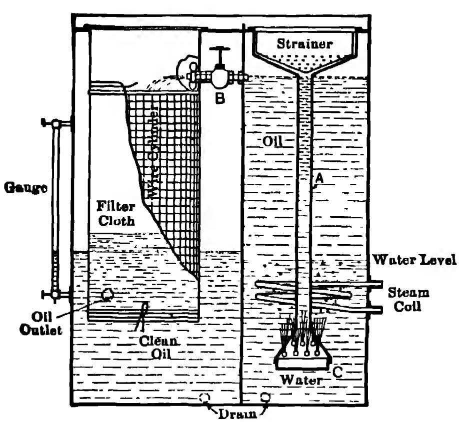


Fig. 615. Typical "Bag-type" Filter.

Figure 616 shows a section through an oil filter in which the filtering medium is a mixture of excelsior, hair felt, and cloth. The general principles are the same as for the smaller design described above, except that the oil is forced through a series of preliminary filters before passing through the bag.

[1] See Report of Prime Movers Committee, N.E.L.A., July, 1925.

For steam turbines the oil from the bearings is usually at such a high temperature that effective separation of water and sediment takes place without heating the oil at the filter. In fact, it is necessary to equip the filter cabinet with water-cooling coils so that the temperature of the oil may be maintained at the desired point.

Plain separating tanks consisting of a single reservoir, with baffles so arranged that the oil must travel vertically downward and then vertically upward a number of times through water at a slow rate, are satisfactory where the soluble sludge content is comparatively low.

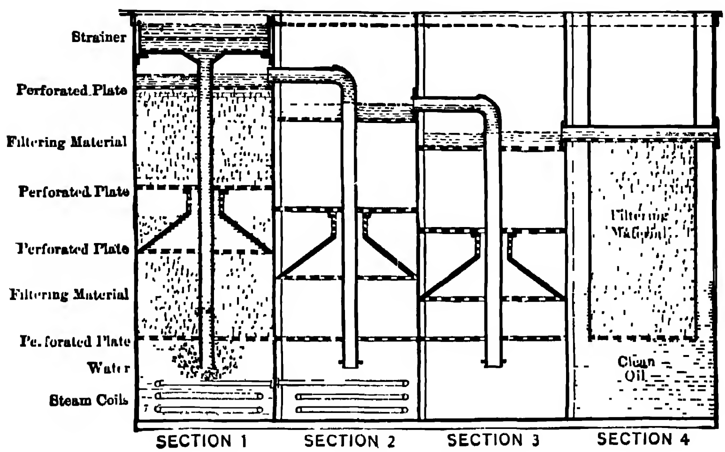


FIG. 616. Turner Oil Filter.

When an excessive amount of acid or alkali is formed in the oil and cannot be removed through the ordinary process of filtration and separation, the oil should be chemically treated to neutralize the product.

Mechanical Separation. — Centrifugal separators operating on the principle of the well-known cream separator are suitable for removal of water and solid materials from oil, but do not in one operation eliminate all the undesirable contaminating substances, especially those that have a specific gravity close to that of the oil itself. They do not remove acids or alkalies, or the soluble sludge, and they offer opportunities for the oil to mix with air. Centrifugal filters similar in principle to the centrifugal separators have been built, but so far have not been able to produce the quality of clarification that is obtained with the series filter. Centrifugal separators are frequently used in conjunction with settling tanks and filters, either for preliminary purification or for removing water after the oil has passed through the filters.

333. Turbine Oil-purification Systems. — There are at present four general methods of oil purification: (1) the **continuous filtration,** (2) the **continuous by-pass,** (3) the **batch,** and (4) the combined **continuous by-pass batch system.**

Continuous Filtration System. — In this system the entire volume of oil is filtered each time it is pumped by the oil pumps of the circulating system. For very small turbines, where only a small quantity of oil is circulated, this method is very effective and is now standard practice. The capacity of the oil tank or tanks should be such that it will take at least 5 min. to circulate a quantity of oil equal to the tank capacity. For larger turbines, however, the space occupied by the filter and its accessories is objectionable and the first cost is excessive.

Continuous By-pass System. — This system differs from "continuous filtration" in that only 10 to 20 per cent of the lubricant contained in the turbine oil reservoir is continuously by-passed through the filtration system. The continuous by-pass system is shown diagrammatically in Fig. 617. A sight overflow *A*, vented to prevent siphoning, is connected to the turbine so that its overflow pipe corresponds with the desired level of oil in the turbine reservoir, which level is thereafter automatically maintained at all times. This arrangement assures that the turbine oil reservoir will always be at the proper level, since the quantity of dirty oil overflowing to the filter is balanced by the amount of clean oil delivered to the reservoir by the filter pump. Accidental stoppage of the filter pump does not prevent continuous operation of the turbine, because the vented overflow makes drainage

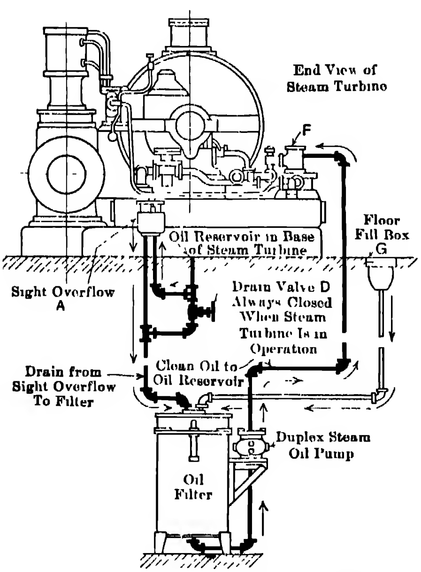


FIG. 617. Continuous By-pass System.

of the turbine oil reservoir impossible. Centrifugal separators and separating tanks are sometimes used in place of the bag filter. This system does not lend itself, however, to the removal of soluble sludge at turbine operating temperatures; but, by the use of the proper oil and an efficient filtration system, the water and air may be continuously removed so that there is little opportunity for this formation to take place. New turbines equipped with this system are reported to have been in opera-

tion for periods up to 2 1/2 years with little deterioration in the lubricating value of the oil. Small quantities of make-up oil are added periodically to replace losses.

Periodical Batch. — In this system the entire charge of the lubricating system of the unit is drained at definite intervals and a fresh supply is introduced. The used oil is submitted to the customary filtration and purification process and stored until the next replacement takes place. The chief objections to this system are that it requires keeping in stock a large spare supply of oil, complicating the storage facilities, and necessitates shutting down the unit each time the oil is charged.

Continuous By-pass Batch System. — Manufacturers of this system, which is a combination of the continuous by-pass and the batch systems, claim that it is possible to remove completely both soluble and insoluble sludge.

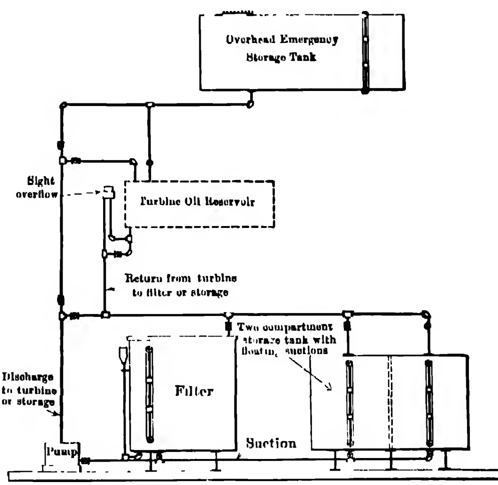


FIG. 618. Continuous By-pass Batch System.

Referring to Fig. 618 it will be noted that the piping is arranged so as to place the usual type of bag filter in the circuit of a continuous by-pass system and so that oil can also be overflowed from the turbine reservoir into precipitation or storage tanks, whenever desired, while the turbine is in operation. The oil is allowed to settle in these precipitation tanks for a considerable period of time, during which sludge and water settle out. The discharge from the settling tanks is connected inside the tank to a float, thereby keeping the end of the discharge near the oil level and permitting the cleanest oil to be drawn from the tank regardless of the oil level.

Steam Turbine Oil Purification Systems: Report of Prime Movers Committee, N.E.L.A., 1923, Part B, p. 320; 1922, p. 2; 1921, p. 8.

The Causes of Trouble in Steam Turbine Lubrication and Remedy: Power, Aug. 17, 1920, p. 244.

Deterioration of Turbine Oils in Use: Power, Oct. 30, 1923, p. 707.

Effect of High Temperatures on Lubricating Oil in Circulating Systems: Power, May 16, 1922, p. 781.

Preventing Emulsification in Oil-Circulating Systems: Power, Nov. 9, 1920, p. 740.

Maintaining Quality of Steam Turbine Oils in Service: Power, Jan. 22, 1924, p. 125.

CHAPTER XVIII

TESTING AND MEASURING APPARATUS

334. General. — The importance of maintaining a system of records is discussed in paragraph 354. The various items which may be recorded and the instruments and appliances used in this connection are outlined in the accompanying chart. No hard and fast rule can be laid down stating what instruments must and must not be used, since the problem is one influenced very much by local conditions. It is better to install too few instruments than to install more than can be taken care of by the plant organization. There are certain instruments which all plants, regardless of size, should have installed. The essential instruments recommended by the N.E.L.A. Committee on Prime Movers are as follows: Steam-pressure gages on boilers, header, and turbine throttle; feedwater-pressure gages; mercury column on turbine exhaust; thermometers for feedwater, steam (if superheated), and bearing oil temperatures. In addition to these there should be the usual electrical instruments. There should also be means for determining the net kw-hr. delivered by the station and for measuring coal. In large stations a full complement of indicating, recording, and integrating instruments may prove to be a good investment if intelligently and closely studied with a view to locating and eliminating unnecessary losses. The instruments should be inspected and calibrated at intervals, since many of them are delicately constructed and are apt to become inaccurate after a few months' service. Steam gages, thermometers, and pyrometers, and particularly piston water meters are subject to appreciable error after considerable use. Voltmeters, ammeters, and other switchboard instruments are easily deranged, especially when subjected to continuous vibration or to high temperature.

TESTING AND MEASURING APPARATUS

Steam Plant

Weights	Fuel	Platform scales, indicating and recording.	
		Suspension hoppers, indicating and recording.	
		Coal meters, integrating.	
	Fluid	Platform scales and tanks.	
		Volumetric	
		Current	Indicating, recording and integrating.
		Area	
		Dynamic	
		Weir	
		Force	
		Thermal	

Pressures	High	Bourdon gage, indicating and recording.
		Manometers, mercurial, indicating.
	Low	Manometers — mercurial, indicating, and recording.
		Manometers — water, indicating, and recording.
		Diaphragms, indicating and recording.
Temperatures	−330 to +1300° F.	Electric resistance thermometer.
	−40 to +800° F.	Mercurial glass thermometers.
	−60 to +200° F.	Liquid pressure thermometers.
	+50 to +400° F.	Volatile-liquid pressure thermometers.
	+120 to +1000° F.	Gas pressure thermometers.
	0 to +1600° F.	Base metal thermocouples.
	+800 to +2900° F.	Rare metal thermocouples.
	Over 2900° F.	Optical and radiation pyrometers.
Power	Indicated	Indicators, hand-manipulated.
		Indicators, continuous recording.
	Developed	Rope brake
		Prony brake
		Absorption dynamometers.
		Electric generator.
Flue-gas analysis	Orsat apparatus.	
	Hand analyzers.	
	Recorders.	
	Caustic.	
	Electrical.	
Moisture	In air	Hygrometer, indicating and recording.
	In steam	Calorimeters: Separating.
		Calorimeters: Throttling.
Fuel analysis	Coal calorimeters	Mahler bomb.
		Parr
	Gas calorimeter	Junker.

Electrical Plant

Voltage	Voltmeters, A. C. and D. C., indicating and recording.
Current	Ammeters, A. C. and D. C., indicating and recording.
Output	Wattmeters, A. C. and D. C., integrating and recording.
Power factor	Power factor meters, A. C. only, indicating and recording.
Frequency	Frequency meter, A. C. only, indicating.
Synchronism	Synchronizers, A. C. only, indicating.

The N.E.L.A. Committee on Prime Movers offers the following list of instruments, arranged roughly in order of importance, which may be expected in the average large central station:

For Station Operators

Turbine Room Instruments

(*a*) Indicating steam gage at throttle and other points, depending upon the type of turbine or engine; mercury column; thermometers for steam temperature (if superheated) and bearing oil; indicating wattmeter; barometer (Aneroid type may be used if frequently checked).

(*b*) Thermometers for circulating water inlet and discharge and condensate.

(*c*) Device for measuring air leakage where possible.

(*d*) Steam-flow meter or condensate meter.

(*e*) Device for measuring tube leakage.

Boiler Room Instruments

(*a*) Steam gage on each boiler (where uniform steam pressure is important and difficult to maintain, a large master steam gage on header is desirable).

(*b*) Feedwater pressure gage.

(*c*) Draft gages at uptake, over fire and air-pressure gages in air duct and at those points of the stoker required by that particular type.

(*d*) Steam-flow meter on each boiler.

(*e*) Air-flow gages on CO_2 recorder.

(*f*) Load indicator or total flow steam meter.

(*g*) Feedwater thermometer.

For Checking Operation

(*a*) Kw-hr. meters and coal scales; steam pressure (and temperature) recorders.

(*b*) Boiler-feed temperature recorder.

(*c*) Vacuum recorder; steam-flow meter or feedwater meter or both; blow-down meter (where this is impractical, a recording thermometer placed in a blow-down line indicates frequency of blowing and leakage of blow-down valves).

(*d*) Feedwater pressure recorder.

(*e*) CO_2 recorder; flue-temperature recorder.

335. Fuel Measurements. — In many small plants using truck or wagon delivery, the delivery tickets of the coal dealer are depended upon for the weight of coal used, no attempt being made to determine the evaporative value; and the economy of the plant is judged by the size of the coal bill. In such cases a considerable saving may be effected by keeping a daily record covering at least the coal and water consumption. The coal can be conveniently weighed on ordinary platform scales or by means of calibrated containers. In large central stations, when the quantities involved are very large, the coal is frequently weighed in the cars at the plant and no attempt is made to determine the amount used by the individual boilers. In many of the modern plants the weight of coal is determined by suspended weighing hoppers, which may be stationary, as in Fig. 193, or mounted on a traveling truck, as in Fig. 192. The scales of such devices are made indicating, recording, integrating, or a combination of the three, the last costing but little more than the simple indicating or recording devices.

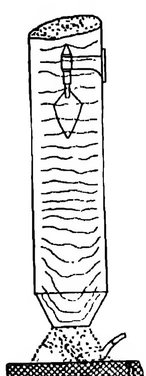


Fig. 619. Bailey Coal Meter.

A simple and inexpensive coal meter is illustrated in Fig. 619. It consists essentially of a helical vane placed in a cylindrical conduit. The movement of the coal causes the vane to rotate, and the number of revolutions is a measure of the volume of fuel passing.

This motion is transferred through flexible shafting to a counter located at any convenient point. The manufacturers guarantee accuracy within 1 to 4 per cent of scale weight, when the meter is installed in a vertical pipe sufficiently large in proportion to the coarseness of the coal and sufficiently long to insure uniform distribution of coal throughout the pipe.

Figure 620 gives a diagrammatic arrangement of the principal elements in the " **Republic** " coal meter as applied to chain grate stokers. For a given depth of fire, the speed of the grate is a function of the volume passing into the combustion chamber. The speed of the grate is trans-

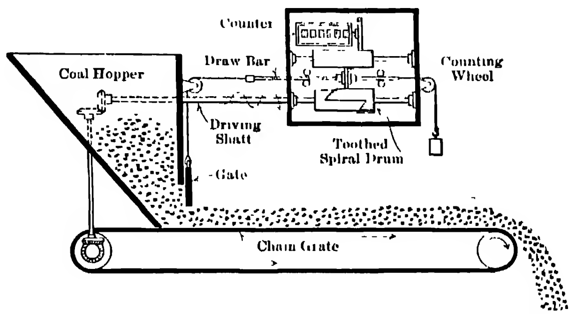


FIG. 620. Principle of the "Republic" Coal Meter.

mitted through suitable linkage to the counter, so that the total number of revolutions over a given period of time is a fairly accurate index of the volume of fuel fed into the furnace during that time. Variations in height of gate are automatically compensated for by a toothed spiral drum, so that no corrections of the meter readings are necessary. In the commercial design, the toothed spiral drum is replaced by a simpler mechanism operating on the same principle, and the register is connected to the stoker shaft by means of a flexible coupling. The manufacturers of the " Lea " coal meter, which is the English design of this device, guarantee an accuracy within 2 1/2 per cent of the true volume for all grades of coal under average working conditions. By checking volume against weight for any given size and grade of coal, the meter readings may be converted into weight by the use of a single constant. This device is also equipped with a tachometer graduated to give the rate of combustion as well as the total over an elapsed period of time.

Volumetric measurements of the coal fed by underfeed stokers may be recorded by calibrating the displacement of the feeder rams and totaling the number of strokes with any type of continuous counter. With very wet fine-sized coal there is danger from arching, and the accuracy of the

displacement of the rams is not a true measure of the volume of coal fed into the furnace. For relation between weight and volume for various coals, see paragraph 118.

The weight of oil fuel fed to the furnace is obtained from the readings of suitable fluid meters, and that of powdered fuels by weighing the product in closed tanks.

Coal Weighing Devices and Meters: Power Plant Engrg., Jan. 1, 1920, p. 70.

336. Measurement of the Flow of Fluids. — The various devices used in this connection have been classified by the A.S.M.E. Special Research Committee on Fluid Meters as follows:

Division	*Class*	*Type*	*Division*	*Class*	*Type*
Positive	Weighing	Weighers	Inferential	Dynamic	Venturi
		Tilting traps			Flow nozzle
	Volumetric	Tank			Orifice
		Piston			Pitot
		Disk			Centrifugal Friction
		Rotary		Weir	Square notch
		Bellows wet-drum			Triangular notch
Inferential	Current	Propeller			Special notch
		Turbine		Force	Hydrometric pendulum
		Helical			Vane
	Area	Gate		Thermal	Electric
		Orifice and plug			
		Cone and disk			
		Cylinder and plunger			

Meters of the weighing, volumetric, and current class give *total quantity* directly regardless of the rate of flow, while those of the dynamic, weir, force, and thermal class give *rates of flow*. When the latter are designed to give total quantity, a mechanism involving a time element must be added.

Most fluid meters consist of two distinct parts each of which has different functions to perform. The first is the **primary element,** which is in contact with the fluid and is acted on directly by it; the other is the **secondary element** which translates the action of the fluid on the primary elements into volumes, weights, or rates of flow and indicates or records the result.

The term **positive** is used to designate those meters through which the fluid passes in successive isolated quantities, either by weight or volume. These quantities are separated from the main stream and isolated by alternately filling and emptying containers of known capacity, and no fluid can pass through such a meter without actuating the device. The secondary element of a positive meter consists of a counter with suitably

graduated dials for indicating the total quantity that has passed through the meter up to the time of reading.

The term **inferential** applies to all meters through which the fluid passes in a continuous stream. The functioning of the primary element depends upon some property of the fluid other than volume or weight, and the secondary element embodies some device which draws the necessary inferences automatically so that the observer may read the results from a dial. For a detailed discussion of the various types of measuring devices outlined above, consult "First Report of A.S.M.E., Special Research Committee on Fluid Meters" published by the Society in 1922.

Measuring Flow of Fluids Power, Mar. 30, 1920, p. 503.
The Salt Velocity Method of Water Measurement: Mech. Engrg., Jan. 24, 1924, p. 13.

337. Measurements by Weight. — Whenever it is desired to calculate the amount of heat absorbed or given up by a liquid, it is ultimately necessary to record the quantity of liquid involved in terms of weight. Any means of determining the weight directly is usually more accurate than a method which consists of measuring the quantity volumetrically and then transferring to a weight basis, since the weight is independent of other physical properties. For this reason, when extreme accuracy is necessary, the liquid is weighed directly, usually by the use of two or more tanks resting upon scales filled and emptied alternately. Where the quantities to be measured are comparatively small and the test is of short duration, it is customary to empty and fill the tanks and effect the weighing by hand. Where large quantities are involved and continuous observations are to be made over a long period of time, this method is impractical because of the cost of attendance and the bulk of the weighing apparatus. When hot liquids are measured in this manner, evaporation may also cause an appreciable error. The principal use of the weighing tank method has been for making boiler tests and for calibration purposes. For specific instructions concerning the weighing of feedwater by means of tanks and scales, consult "Rules for Conducting Boiler Trials," A.S.M.E. Code of 1925.

338. Volumetric Meters. – In this class of positive meters, volumes and not weights are measured, though the scale may be graduated to read weight. For constant density the weight readings are fully as accurate as the volume readings, but for varying densities corrections must be made. In some designs these corrections are automatically made by the meter mechanism. The volumetric meters of the positive type most commonly found in power plant practice are the **tank,** of which the Wilcox, Fig. 621, is an example; the **piston,** Fig. 622; the **disc,** of which the Nash, Fig. 623, is a typical example; and the **rotary,** Fig. 624, which

is usually limited to small sizes. Piston, disc, and rotary meters are usually placed on the pressure side of the boiler-feed pump (the condensation type of rotary meter excepted) since a pressure difference is necessary to actuate the mechanism, while tank meters are generally placed on the suction side. The former take up considerably less room than the latter, but are subject to wear with consequent leakage and error in readings. In many plants where piston, disc, and rotary meters are installed, the meter is by-passed so that it can be operated for short periods as well as " cut out " for repairs.

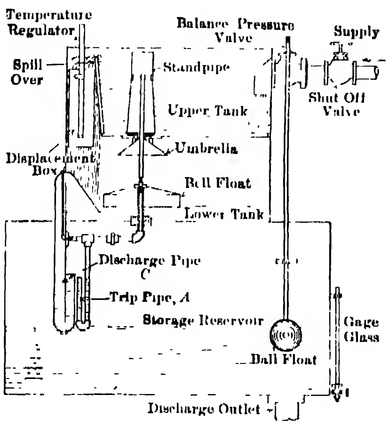


FIG. 621. Typical Tank Weigher (Wilcox).

Figure 621 shows a diagrammatic outline of the **" Wilcox Automatic Water Weigher,"** illustrating a typical volumetric meter of the "tank" class. The device consists of a metal tank divided into an upper and lower compartment by a horizontal partition and a float-controlled storage reservoir. The water enters the upper compartment, passes to the lower, in which its volume is measured, and then out through the U-shaped discharge pipe. The operation, beginning with both compartments empty,

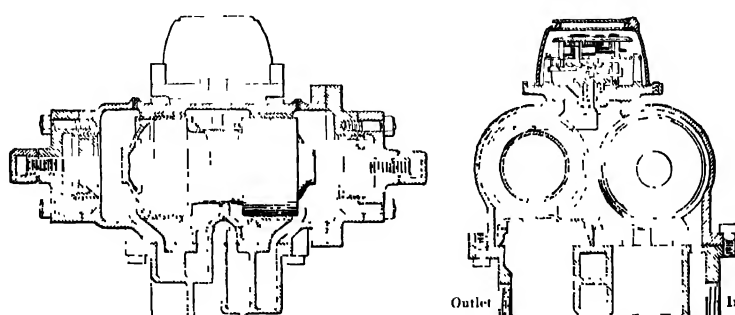


FIG. 622. Typical Piston Meter (Worthington).

is as follows: Water enters the upper compartment through the inlet pipe and rises to the top of the standpipe. (The latter is open at the top and bottom and is rigidly connected to the bell float, but when in its lowest position it is held against its seat by weight of the bell float.) Further admission of water causes it to overflow into and through the

standpipe into the lower compartment. The water, rising in the lower compartment, seals the lower edge of the bell float and entraps a volume of air under the bell. Further rise compresses the air under the float, in leg C of the discharge pipe and in leg A of the trip pipe AB. This compression causes the float to rise to its highest position and raises the standpipe from its seat, permitting the water in the upper chamber to pour into the lower vessel. Compression of air continues until the pressure becomes great enough to break the seal in the trip pipe. This action immediately reduces the pressure below the float, permits the latter to descend, sealing the upper chamber against further discharge, and allows the water in the lower compartment to siphon out through the discharge pipe. The number of discharges is recorded mechanically. The ball float and balance pressure valve are in operation only when the "weigher" is not working at its full capacity. This particular design is made in one size only, maximum capacity 35,000 lb. per hr.

Figure 622 shows sections through a **duplex-piston** meter which is essentially the water end of a duplex double-acting pump having the cross-over valve motion at the bottom. The moving parts consist of two plungers and two side valves with a lever which conveys the motion of the plunger to the recording mechanism. By means of adjustable tappets at the ends of the cylinder, the length of the plunger stroke and consequently the displacement per register may be altered. This provides means for calibrating the meter for any service. These meters are not suitable for capacities over 350 gal. per min.

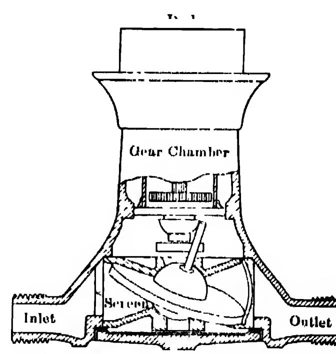


FIG. 623. Typical Disc Meter (Nash).

The **disc meter,** Fig. 623, consists of a measuring chamber which is divided into two compartments by a disc revolving about a spherical bearing. One compartment is always filling while the other is emptying and thus an unbalanced fluid pressure moves the disc. Each revolution of the disc displaces a definite volume of fluid. These meters are not constructed for capacities over 500 gal. per min.

Figure 624 illustrates the principles of the **Cadillac condensation** meter, illustrating the rotary type. Its mechanism consists primarily of a cylindrical copper drum divided into six scroll-shaped compartments, together with a case and an integrator. The fluid is admitted by means of a spout introduced axially at the center of the drum, and extending throughout its entire length is the inlet opening by which the water passes each com-

partment successively as the drum rotates. Starting with compartment 1, directly beneath the spout, the incoming fluid fills it, and the peculiar shape of the compartment causes the greater portion of the fluid to flow to one side of the perpendicular side line of the drum. By seeking to find the center of gravity, the water rotates the drum, until finally, when compartment 1 is full, the weight and location of the fluid have drawn compartment 2 beneath the spout. This second compartment fills in its turn and draws compartment 3 into place, and the fluid in compartment 1 now begins to overflow at its outer opening in the perimeter of the drum. Compartment 1 does not commence to discharge until compartment 3 is partly filled. All compartments are alike and each receives and discharges the same volume of fluid. These meters are intended for capacities not exceeding 35 gal. per min.

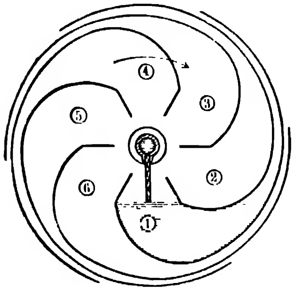


FIG. 624. Principle of the Cadillac Condensation Meter.

339. Dynamic or Velocity Meters. — In this class of meters the stream of fluid creates a difference of pressure, or a differential head, through the primary device, this head depending upon the velocity and density of the fluid. The Venturi tube (Fig. 625), flow nozzle (Fig. 635), orifice (Fig. 623), hyperbolic elbow, and Pitot tube (Fig. 631), are the means commonly employed for creating the differential head.

The relation between pressure and velocity in all types of dynamic or velocity meters is expressed by the basic law

$$V = C\sqrt{h} \tag{282}$$

in which

V = velocity, ft. per sec.,
h = head of fluid producing flow, ft.,
C = experimentally determined coefficient which varies with means of producing the differential head and other construction factors.

Meters of this class are suitable for measuring the flow of gases and vapors as well as that of liquids. The Venturi tube and hyperbolic elbow necessitate cutting the pipe line for their introduction; nozzles and orifices are usually designed so that they may be slipped in between the pipe flanges; but the Pitot tube may be inserted in all but very thin and small diameter tubes by merely drilling a hole in the pipe at some convenient point. Dynamic meters are applicable only to fluids flowing in closed

conduits and where the velocity is sufficient to create an appreciable differential head. In the commercial designs of this class of meters, the simplest and most inexpensive device for indicating the rate of flow is the manometer in any of its forms. Total quantities are obtained by taking periodic readings, averaging, and multiplying by the total time. The difference of head, and hence the rate of flow, may also be transmitted to indicating and recording points by the displacement of floats resting upon the surface of the liquid in the manometer. For recording total quantities the time element must be introduced. The basic principle of the totalizing or integrating mechanism is illustrated in Fig. 626.

Among the well-known dynamic meters used in power plants for measuring the flow of fluids in pipes, may be mentioned the Builders' Iron Foundry Company's Venturi; the " Republic Flow " (R.F.M.), " General Electric " (G-E), " Cochrane," and " Bailey," involving the use of orifices, flow nozzles, or Pitot tubes; the " Hyperbo-Electric," employing the hyperbolic elbow; the " St. John's," utilizing an orifice and plug; and the " Simplex," which has a Pitot tube for its primary element.

For satisfactory operation, the primary elements of all meters of the dynamic class should have a straight run of pipe approximately 20 diameters in length ahead of them, and the water should be comparatively free from dirt or scale which may plug up the various openings or cause corrosion of the nozzles or tubes. Violent fluctuations and pulsating flows also affect the accuracy of the readings.

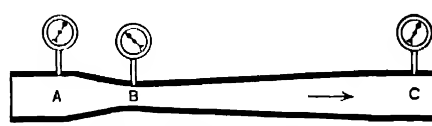


Fig. 625. Diagram of Venturi Tube.

The **Venturi tube** consists of a tube of circular cross section, shaped as shown in Fig. 625. Starting at the upstream flange, there is first a short cylindrical piston, machined inside, which is substantially a continuation of the pipe line. In this piston several small holes lead into a piezometer ring, so that a connection may be made for measuring the static pressure before it enters the construction. An entrance cone of about 21 deg. total angle connects the short straight section with a short cylindrical **throat**. The latter is machined and provided with side holes leading to a piezometer ring for measuring the static pressure at this point. The end of the throat leads into the exit cone or diffuser, which has a total angle of about 5 to 7 deg. This terminates in the downstream flange for connecting the Venturi to the following pipe line. Venturi tubes under 2 in. in diameter are usually made entirely of bronze and finished inside throughout the entire length, while the larger sizes are generally of cast iron with the throat and the straight entrance lined with bronze and machined to a smooth finish.

The so-called theoretical equation of the Venturi meter for the flow of water is

$$Q = M\sqrt{2gh} \tag{283}$$

in which

Q = rate of discharge, cu. ft. per sec.,
g = acceleration of gravity = 32.2 approx. for average purposes,
h = difference in head between the static pressure at the two piezometer connections, ft. of water,
$M = A/(r^4 - 1)^{\frac{1}{2}}$, where A = area of the entrance section at the up-stream pressure connection, sq. ft., and r = ratio of entrance to throat diameter. M is therefore a constant for a given size and design of meter.

The actual rate of discharge may be obtained from equation (283) by multiplying the calculated results by a coefficient C, which varies with the throat speed, diameter of the pipe, viscosity of the fluid, and other factors. In practice the numerical value of C is determined from experimental tests. The commercial type of Venturi for measuring the flow of water has a coefficient of 0.96 or over for all rates of flow within its designated range of capacity.

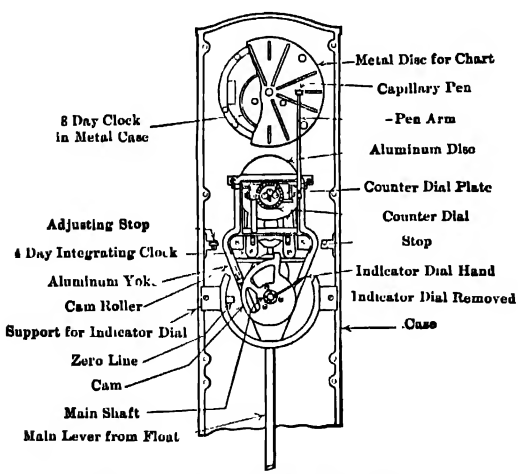


FIG. 626. Interior of Register Mechanism.

Figure 626 shows the interior of the registering mechanism for indicating, recording, and totalizing the rate of flow. Two small tubes connected with the inlet and throat of the tube transmit the difference of pressure at these points to the meter or registering device. At the back of the register these pipes enter two large vertical wells, connected at the bottom by a small pipe, one well being subjected to the inlet pressure and the other to the throat pressure of the tube. In each well is a heavy metal float resting upon mercury, which flows from one well to the other in direct proportion to the difference of the two pressures, causing one float to rise as the other descends, the movement being transferred through rack and spur gearing to the indicator

dial and shaft. A cam on this shaft controls the position of the pen on the chart and also the degree of movement of the counter dial figures. The operation of the registering mechanism is illustrated in Fig. 626.

Venturi Meter: Report of A.S M E. Special Research Committee on Fluid Meters, 1922; Trans. A.S.C.E., Vol. 17, p. 252; Trans. Inst. C.E., Vol 199, 1914–15, Part 1; Proc. Roy. Soc., Vol. A83, 1910, p. 366; Power, Jan. 23, 1912, p. 102.

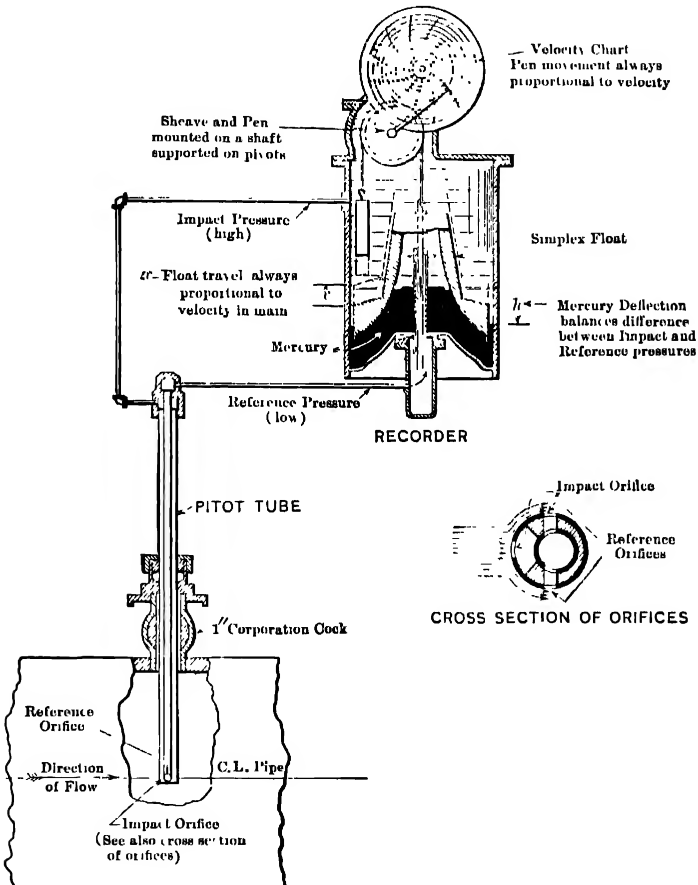


FIG. 627. Principles of "Simplex" Pitot-tube Meter.

Figure 627 shows the principles of operation of the **"Simplex" Pitot meter** with recorder attachment. A unique feature of this meter is the **Ledoux** bell, the use of which permits the direct indication or recording of the velocity or velocity head. When equal pressures are delivered from the two connections of the Pitot tube, the bell, under the influence

of its counterweight and its weight of water, stands in the mercury at the lower level of the curved interior. When subjected to a differential pressure, it moves downwards, displacing an amount of mercury equal to that which is lifted upward into the float, until equilibrium has been established. The downward motion is proportional to the square root of the mercury displacement, and, since the latter is proportional to the square of the velocity, it is evident that the movement of the float is directly proportional to the velocity or rate of flow.

For a description of commercial meters involving the use of orifices, flow nozzles, and hyperbolic elbows, see paragraph 312.

The appropriations of the great majority of small steam power plants do not permit of the installation of tank meters, Venturi meters, or other forms of commercial appliances for measuring the weight of water fed to the boilers. For use in such cases, an inexpensive and fairly accurate

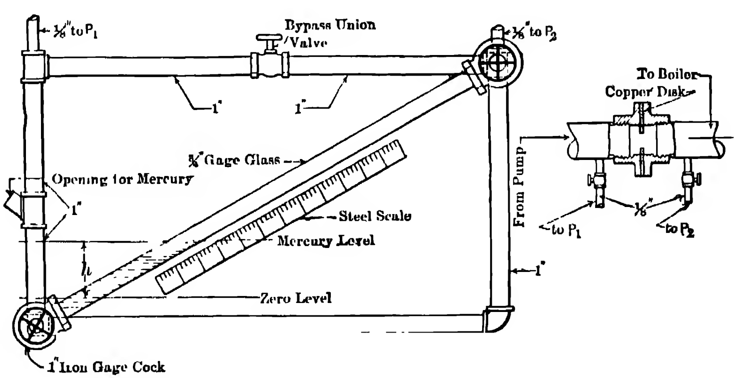


FIG. 628. Simple Indicating Water-meter. Orifice Type.

indicating meter may be constructed of ordinary pipe fittings, as illustrated in Fig. 628. A thin metal diaphragm with circular orifice is inserted on the pressure side of the feed pump, and the pressure drop across the orifice is measured by an inclined mercury manometer. The height of mercury h is an indication of the rate of flow. By calibrating the manometer against tank measurements, the readings of the mercury column may be graduated to read directly in lb. per hr. If means of calibration are not available, the weight of discharge may be approximated from the formula

$$W = 1120\, a \sqrt{hd}, \tag{284}$$

in which

W = weight flowing, lb. per hr.,
a = area of the orifice, sq. in.,

h = vertical height of mercury column, in.,
d = density of the water, lb. per cu. ft.

For a fairly continuous flow and pressure drop corresponding to 3 in. of mercury or more, this simple device gives results agreeing within 4 per cent of tank weights; but for widely fluctuating flow and small pressure drops the error may be considerably more.

340. Weir Measurements. — For measuring the flow of streams or of large quantities of water in open conduits, the weir offers an accurate means of determining the rate of flow. For high heads, such as may be found in the measurement of large streams, the rectangular notch is commonly used; but where the heads are comparatively low, as in most power plant service, the triangular notch is the more reliable. In the weir meter the area of the stream and the head are variable, but not independently. The variable area bears a definite relation to the velocity and is indicated by the height of the level of the liquid measured over a horizontal plane at the base. For the ordinary rectangular notch with two contractions, and heads ranging from 3 in. to 2 ft., it has been found by experiment that

$$Q = 3.33(b - 0.2\,h)h^{\frac{3}{2}} \tag{285}$$

in which

Q = cu. ft. per sec.,
b = length of the weir, ft.,
h = height of liquid passing over the weir, ft.

For the triangular notch with 90-deg. angle, the rate of flow is

$$Q = 0.305\,h^{\frac{5}{2}} \tag{285a}$$

Weir meters are also used extensively for the measurement of feedwater, condensate, and blow-off discharge. Among the popular designs may be mentioned the Lea V-notch, Cochrane, Hoppes, Bailey, and Webster.

Figure 629 shows the general principles of the Yarnall-Waring Company's **Lea V-notch** recording liquid meter. The head of water flowing over the notch is measured by means of a float operating in a still-water chamber out of the path of flow. Movement of the float is transmitted to the indicating and recording apparatus through the agency of a small spindle. The upper end of the spindle indicates on a graduated scale the depth of water over the weir. The movement of the spindle is also transmitted through a gear to a rotating drum. This drum is grooved so that a slider bar engaging the groove actuates a recording pen in direct proportion to the rate of flow. An integrating or totalizing mechanism may be readily attached to the slider bar. Variations in density due to temper-

ature are automatically compensated for by the float, since the depth of immersion increases as the density decreases and vice versa. Lea V-notch meters are available in a number of sizes and designs, with unit capacities up to a maximum of 1,000,000 lb. per hr., and readings when carefully calibrated are guaranteed to be accurate within 1 1/2 per cent of actual tank weight.

Weir Meters: Power, May 1, 1917, p. 582; Trans. Neb. Soc. Engrs., Vol. 1, No. 1; Eng. News, 1914, Vol. 2, p. 277; Proc. Am. Water Wks. Ass'n, 1912; Mech. Engrg., Feb., 1920, p. 83.

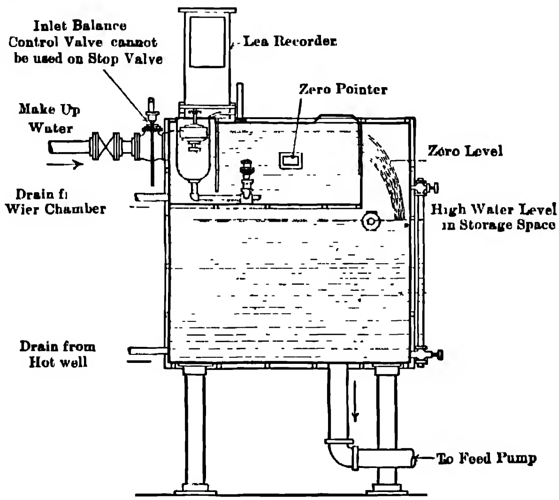


FIG. 629. Typical Weir Meter (Lea V-notch)

341. Steam Measurements. — The quantity of steam passing into any system may be determined (1) by collecting and weighing the condensate at intervals or by passing it through suitable liquid meters, and (2) by measuring the rate of flow of the steam itself in pipe lines. The first necessitates the use of surface condensers, unless condensation is effected in the system itself, and consequently has a limited field of application. Any suitable type of water meter may be used for measuring condensate. In large central stations condensation meters of the V-notch type are frequently incorporated in the hotwell chamber of the surface condensers.

342. Flow Meters. — Any of the inferential types of fluid meters outlined in paragraph 336, with the exception of the " V-notch " class, may be arranged to measure the flow of steam in pipes, but the most successful commercial devices are of the " dynamic " and " area " class.

The weight of fluid flowing through an opening may be calculated by the equation

$$W = AyV \qquad (286)$$

in which

W = weight, lb. per sec.,
A = cross-sectional area, sq. ft.,
y = density of the fluid, lb. per cu. ft.,
V = velocity of flow, ft. per sec.

All steam meters for indicating or recording the weight of steam flowing through a pipe are based upon the law expressed in equations (282) and (286). Thus, for steam of constant density, the opening through which it flows may be made constant and the variation in velocity will be an indication of the rate of discharge ("dynamic" class); or the velocity may be held constant and a variation in the amount of opening will be an indication of the weight discharged ("area" class). Unfortunately, the density of steam is seldom constant under commercial conditions, and herein lies the inherent defect of all steam meters which depend for their operation upon a variation in the area of efflux or a variation in velocity. The density of steam is a function of its pressure and quality, and any variation in either will affect the weight of discharge as determined from equation (286). Pressure and temperature variations may be automatically compensated for, but corrections for quality must be made in each specific case.

The average high-grade steam meter is a reliable and accurate means of measuring the flow of steam in straight lengths of pipes, provided the quantity flowing is within the limits specified by the manufacturer, the flow is continuous, the change in the rate of flow is gradual, and the pressure and quality are practically constant. For low capacity, interrupted, or intermittent flow and for sudden variations in pressure or quality, the results are not so reliable and may be in error. The accuracy of all meters, provided they have been correctly calibrated and adjusted, depends largely upon the degree of refinement in reading the indicators and in integrating the charts. The commercial failure of many steam meters is due to the fact that they are not cared for or operated in strict accordance with the principles of design.

Republic Flow Meters. — These meters are of the dynamic class, in which the primary element is a Pitot tube or orifice, and the secondary element a mercury manometer actuating an electric current. The differential pressure produced by the primary device is not used as a motive force for the operation of the mechanism. Its action is that of an electrical manometer. The indications of flow may be obtained on standard electrical switchboard instruments (fixed or portable type), having a high degree of instrumental precision.

The use of this electrical method of measurement eliminates all cams,

floats, levers, gears, and stuffing boxes, removing several of the elements which lead to instrumental errors and derangement in other types. A further feature is the freedom of repetition of meter stations, obtained by wiring additional meter circuits under conditions that would preclude the use of devices in which hydraulic pressure is transmitted from the primary to the secondary device. In other words, there is no reasonable limit to the distance at which the secondary device can be located from the primary.

The fundamental principles are shown diagrammatically in Fig. 630. The meter body, or U-tube, is partly filled with mercury, and is made to balance the dynamic pressure of the flow in the pipe by corresponding rise of mercury in the low-pressure side of the tube. The mercury column forms a part of the electric circuit, as illustrated. The electric circuit contains a fixed external resistance R_1 in series with a variable internal resistance R_2, an electromotive force E, a conductance indicator A, and a conductance integrator W. In the contact chamber C, which forms the low-pressure side of the U-tube, there are a number of conductors of varying lengths placed above the mercury column, and as the mercury rises it makes a contact with one conductor after another. The variable resistance R_2 is subdivided by these conductors into resistance steps corresponding to the varying lengths of the conductors, so that the rise and fall of the mercury column varies the amount of resistance and the corresponding amount of electric conductance in the circuit. The readings of the electrical instruments are controlled solely by the variation in height of the mercury column and are independent of the electromotive force impressed on the circuit or of the current actually passing through the instruments.

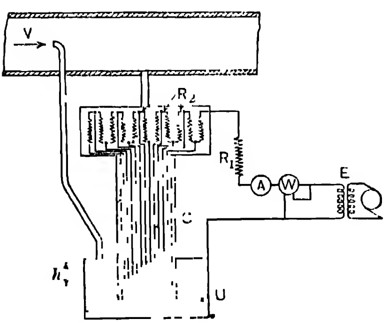


FIG. 630. Principle of the Republic Flow Meter.

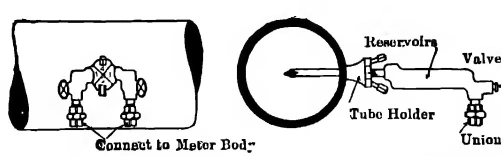


FIG. 631. Pitot Tube. (Republic Flow Meter.)

Figure 631 shows the general assembly of the Pitot tube, which consist of two brass tubes with beveled edges, one facing the flow and the other facing the opposite direction, connected to corresponding reservoirs or condensers through a common tube holder. Pitot tubes are used in situations where the minimum flow in the pipe is sufficient to make a

perceptible difference in pressure, and where the maximum flow does not exceed the range of the mercury column in the meter body.

The orifice plate consists of a monel metal disc with a circular opening in the center. The disc is inserted between two flanges in the line and produces a contracted area in the stream of the flow, which in turn creates the necessary pressure difference for actuating the mercury column. The orifice has the advantage over the Pitot tube in that the pressure difference may be varied within certain limits for a given rate of flow by changing the size of circular opening.

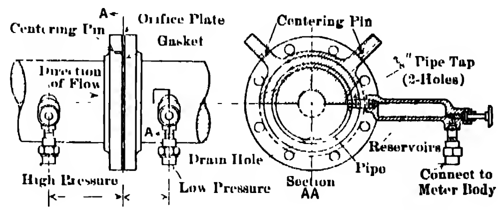


FIG. 632. Orifice Plate. (Republic Flow Meter.)

Figure 633 shows a section through the meter body of the "Republic Flow" meter, which, as will be seen from the illustration, consists of a mercury chamber, a sealing device, and an internal resistance element.

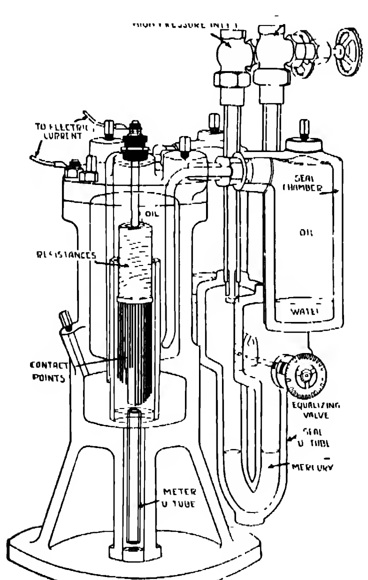


FIG. 633. Meter Body. (Republic Flow Meter.)

General Electric Flow Meters. — The primary element in the "G-E" line of meters is either an orifice tube, flow nozzle or Pitot tube, and the secondary element consists of a specially designed U-tube mercury manometer in which the variations in mercury level are magnetically transmitted to the indicating and recording mechanism. Orifice tubes are used only for pipes under 2 in. in diameter; flow nozzles for pipes 2 in. or more in diameter if the maximum flow is too low to be accurately measured with Pitot tube and in cases where the steam carries boiler compound or foreign matter; and Pitot tubes where it is not convenient to use flow nozzles.

Figure 635 shows the location of the flow nozzle with reference to the pressure attachments, the distances A and B varying with the internal diameter of the pipe. Figure 636 shows the general appearance of the orifice tube and Fig. 637 that of the Pitot tube or nozzle plug. Referring

to Fig. 637, *TT* are the static openings or "trailing set" and dynamic openings or "leading set." *The plug is screwed into*

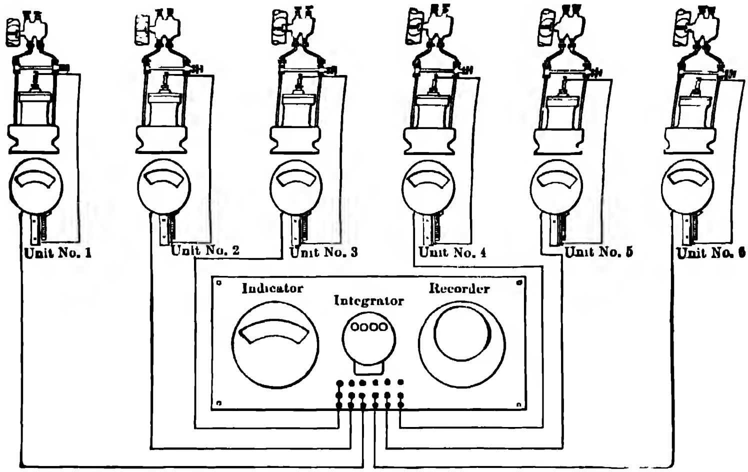


FIG. 634. Typical Arrangement of Republic Flow Meter in a Six-unit Boiler Plant.

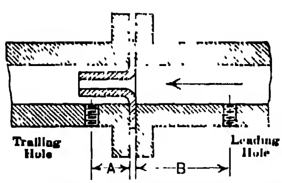


FIG. 635. Typical Flow-nozzle Installation. (G-E Meter.)

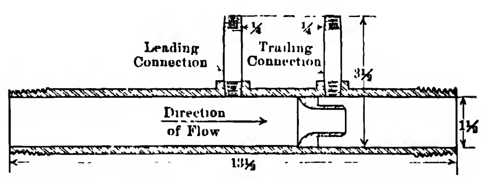


FIG. 636. One-and-one-half-inch Orifice Tube. (G-E Meter.)

with the leading set directly facing the current, and connections to the manometer are made through openings *T* and *L*.

The secondary elements of the mechanically operated device consist of a plain indicating mercury manometer graduated in inches where a portable testing device is desired, and a stationary dial-indicator, a combined indicating and recording, or a combined indicating-recording-integrating device for permanent installation. The general principles of the secondary element of the stationary indicator mechanism are shown in Fig. 638. A small

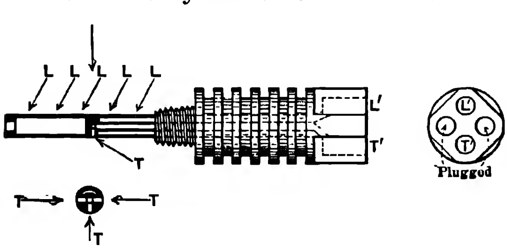


FIG. 637. Nozzle Plug. (G-E Meter.)

float resting on the top of the mercury in one leg of the U-tube is attached to a rack which engages a pinion mounted on a shaft. The shaft on which the pinion is mounted carries a small horseshoe magnet with its pole faces near and parallel to the inside of a copper plug fastened to the body of the meter. A small magnet is mounted on pivot bearings in such a manner that its poles are near and parallel to the outside surface of the copper plug, and its axis of rotation in line with the shaft carrying the magnet inside the case. The indicating needle is attached directly to this magnet. By means of the rack and pinion, the pulley carrying the magnet inside the body is rotated in proportion to the change of level of the mercury. Any motion of this magnet is transmitted magnetically to the outside magnet carrying the indicating needle.

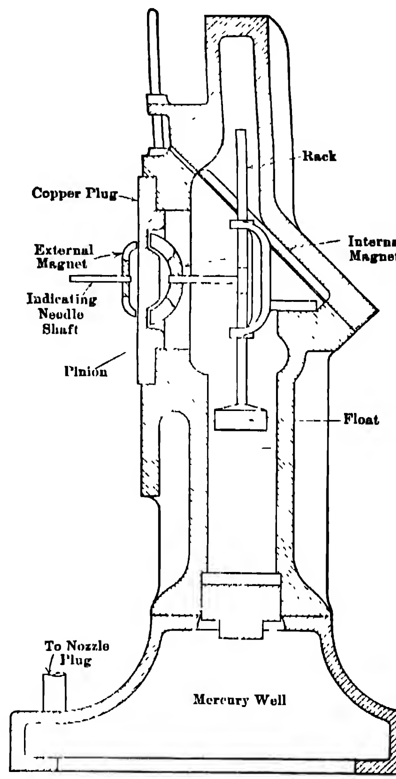


Fig. 638. Meter Body. (G-E Type F-12.)

In the G-E mechanically operated indicating recorder, the dial is a circular chart revolved by a spring-driven or synchronous-motor electric clock, and the positions of the indicator are recorded as a continuous line on the chart. Since the deflection of the indicating needle varies directly with the height of the mercury column and the latter varies with the square of the velocity, it is evident that the ordinates of the chart do not vary directly with the velocity; therefore, in integrating the chart for calculating total flow, a special type of radial planimeter is required.

The meter body of the mechanically operated indicating-recording-integrating instrument differs from the others previously described in that the rack is of circular construction and the magnets are semi-circular in shape. The integrating mechanism consists of a cam mounted on the same shaft and turning through the same angle as the indicating pointer. This cam limits the upward movement of an oscillating arm in proportion to the rate of flow. The arm is oscillated back and forth once every minute by a small clock-operated heart-shaped cam. A ratchet mechan-

ism at the pivot of the arm revolves the registering glass, and dials an amount proportional to the displacement of the arm. The ratchet is arranged so that only the upward movement of the arm actuates the registering gears.

Figure 639 gives a diagrammatic view of a G-E electrically operated flow meter. The primary device consists of the same design of orifice tube or flow nozzle as for the mechanically operated instrument. The secondary device consists of a cast-iron meter body and is essentially a mercury manometer in which the base, or mercury well, forms one leg

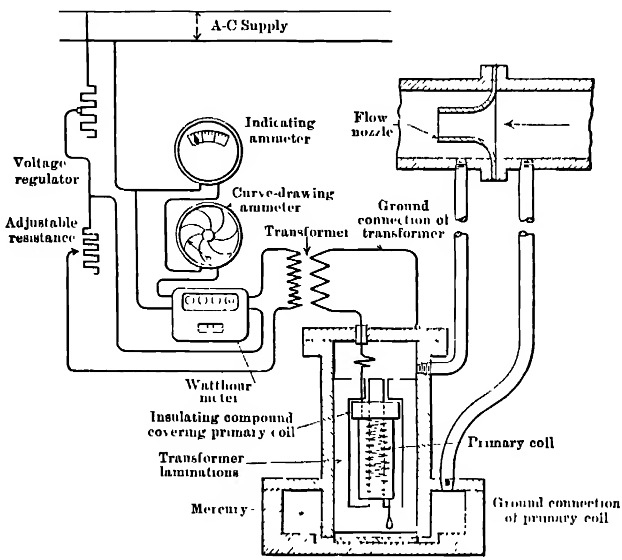


FIG. 639. Diagrammatic View of G-E Electrically Operated Flow Meter.

and the smaller chamber, in which a transformer is inserted, the other. The base, or large leg, of the U-tube is connected to the upstream side of the flow nozzle, and the small, or transformer leg, to the downstream side.

On the outside of the meter body is mounted a small transformer the function of which is to reduce the voltage applied to the internal transformer in the meter body as well as to act as an insulating transformer. One hundred and eight volts, held constant by the voltage regulator, is applied to the primary of this transformer, while the secondary voltage applied to the primary coil of the transformer in the meter body is less than 5 volts.

In series with the primary of the outside transformer are the adjustable line resistance, voltage regulator resistance, and electrical measuring instruments.

When there is no flow of gas or fluid through the main pipe, the electrical instruments indicate the excitation current and the zero readings on the instruments and meters are suppressed, so that zero flow corresponds to this excitation current. When the gas or fluid flows through the pipe there will be a differential pressure produced by the flow nozzle, which causes the mercury in the meter body to rise in the transformer leg and fall in the base until the unbalanced column balances the differential pressure.

As this mercury ring rises around the primary coil of the internal transformer, more and more current is induced in it. This current must be supplied through the primary circuit, the action being similar to pouring mercury in the fiber cup as previously described.

If properly calibrated, the electrical instruments will accurately measure the height of mercury in the small leg of the U-tube containing the internal transformer, which height is a function of the flow of gas or fluid in the pipe.

Bailey Fluid Meters.—Figure 640 shows a section through the meter body of a Bailey fluid meter, the primary element of which is of the thin-plate orifice type, and the secondary element a mercury manometer actuating a "bell" float. The higher pressure is applied at P_1 and the lower pressure at P_2, through small tubes or pipes. The interior of the "bell casing" is subjected to pressure P_2, and the interior of the mercury-sealed "bell" is subjected to the higher pressure P_1. This difference in pressure pushes the bell upward, and, as it rises from the mercury, the change in the buoyant action of the mercury on the walls of the bell balances the force due to the pressure difference. The shape of the bell and thickness of wall and weight are designed so that the lift is directly proportional to the rate of flow through the orifice. This gives a direct-reading chart with uniform graduations and simplifies the design of the integrating mechanism. The principles of the totalizer are shown in Fig. 641. R is a small friction wheel mounted on a shaft (the position of which

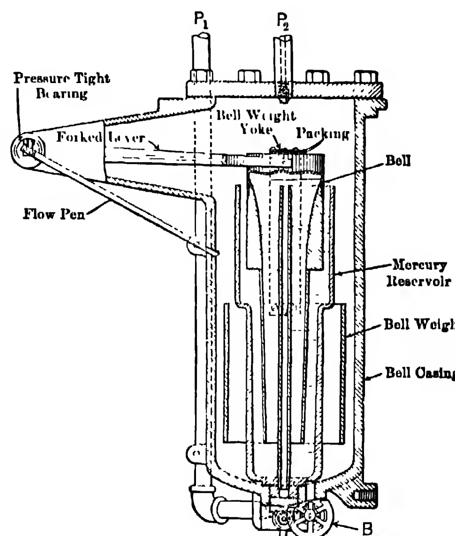


FIG. 640. Section through Meter Body of a Bailey Fluid Meter.

is controlled by the displacement of the bell) and pressing against the surface of a clock-driven friction disc F. The rotations of R are transmitted through suitable gearing to the registering dials D. When there is no flow the friction wheel is at the center of the disc and is at rest. As soon as a flow begins, the displacement of the bell moves the gear away from the center and its speed of rotation is increased directly in proportion to the distance.

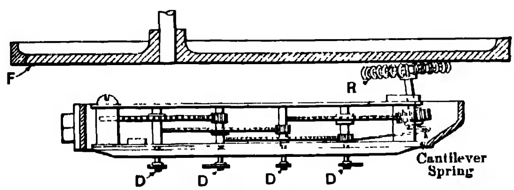


FIG. 641. Totalizing Mechanism. (Bailey Meter.)

Since the movement of the support carrying gear R is directly proportional to the rate of flow, it follows that the number of revolutions is a direct measure of the total quantity for the time element involved.

Cochrane Meter. — Figure 642 gives a diagrammatic outline of the principles involved in the Cochrane flow meter. The primary element is a thin, sharp-edged orifice which can be installed inside the bolts between the flanges in the pipe line. The pressure difference is transmitted through suitable piping to the secondary element, which is essentially a U-tube mercury manometer mounted on knife edges. When there is no flow, the mercury in both legs of the manometer is at the same height and the system is in equilibrium. As soon as there is a flow, the mercury rises in one leg and falls in the other, causing the manometer to tilt in the direction of the greater weight. This motion is transmitted to the indicating dial or recording pen. The tilting is resisted by a cam attached to the U-tube beam and bearing against a flat metallic strap. The cam is so shaped that the tilting is

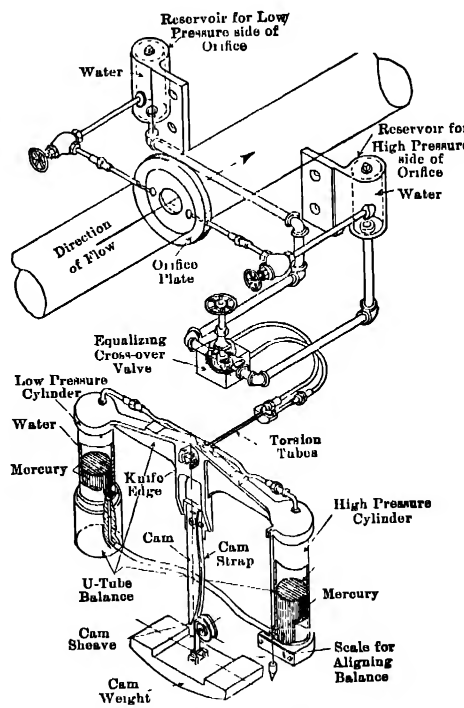


FIG. 642. Principles of the Cochrane Flow Meter.

directly proportional to the rate of flow and permits the use of direct-reading charts with uniform graduations. The entire secondary element is housed in a casing 21 in. in diameter by 8 in. deep. This instrument is made to indicate and record, but no integrating attachment is provided. Corrections for change in density are made by applying "correction factors" to the chart readings.

Hyberbo-Electric Flow Meter. — In this type of dynamic flow meter, the pressure difference is effected by a specially shaped elbow having a rectangular hyperbolic section, preceded by an approach bell and straightening grids. The manufacturers claim that a stream-line flow is produced by the grids and elbow and the relation between pressure and centrifugal force is fixed; and because of the dependence of centrifugal force upon velocity, the relation between pressure and velocity is also fixed. The secondary element is a mercury manometer in which the variations in level of the mercury are transmitted electrically to the indicating, recording, and integrating devices. By means of a Wheatstone bridge and a relay mechanism, the measurements of flow are indicated, recorded, and totalized through the agency of suitable electrical instruments. Connections for pressure and temperature (if the steam is superheated) are automatically compensated for by variations in the resistances of the bridge.

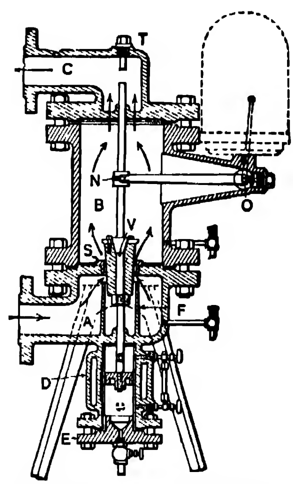


FIG. 643. St. John's Steam Meter.

St. John's Meter. — Figure 643 represents a section through a St. John's steam meter, illustrating a well-known design of area meter of the "orifice and plug" type. This meter was placed on the market about the year 1905 and still finds favor with many engineers. It records the weight of steam passing through the seat of an automatic lifting valve which rises and falls as the demand for steam increases or diminishes.

Referring to the illustration, valve V is weighted so that a pressure in space A 2 lb. greater than that in B is necessary to raise the valve off its seat. This pressure difference is constant for all positions of the valve. The plug is tapered so that the rise of the steam pressure is directly proportional to the volume of steam flowing through the seat. The movement of the valve is transmitted through suitable levers to an indicating dial and a recording pen, so that the instantaneous and continuous rate of flow may be read at a glance. For a given pressure and quality of steam, the indicating dial and chart may be calibrated to read the weight

of discharge directly, corrections being made for variations in pressure and quality. The manufacturers guarantee the readings of the chart to be within 2 per cent of condenser measurements for a total pressure range of 10 lb. from the mean pressure at which the chart is calibrated. The chief drawback to this instrument is inherent in all meters of the direct type in that they are bulky and the steam line must be taken down for the installation.

343. Pressure and Draft Gages. — The **Bourdon** type of gage, either indicating, Fig. 644, or recording, is the most familiar and satisfactory means of measuring pressures up to 1500 lb. per sq. in. or more, although a number of successful high-pressure gages are actuated by diaphragms. The Bourdon type is also available for measuring very low pressures and vacua, but the mercurial vacuum gage has the advantage of greater accuracy and is not subject to derangement. High-pressure gages of the Bourdon, diaphragm, or spring type, should be frequently standardized since they are subject to error through use.

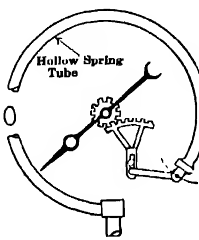


FIG. 644. Principles of the Bourdon Pressure Gage.

For furnace draft and other low-pressure measurements, there are a number of successful instruments on the market which depend for their action upon gravity, sylphon bellows, weighted diaphragms, and floats. The simplest and most inexpensive type of indicating device for low pressures or low-pressure differentials is some form of liquid manometer. These manometers are available in a wide range of sizes and designs, from a plain vertical **U-tube,** Fig. 645, up to an instrument 30 in. in length and reading directly to 1/10-in. increments, and to inclined gages, Fig. 646, giving total pressures of 1/2 in. of water and graduated to read to 0.01-in. increments. Mercury, water, and oil are the liquids usually employed. The sensitiveness of these liquids to pressure changes are as follows (temperature of liquid 62 deg. fahr.): mercury 1, distilled water 13.6, 120-deg. water-white kerosene 17. The height of a vertical column of these liquids (temperature 62 deg. fahr.) which will balance a pressure of 1 lb. per sq. in. is as follows: mercury 2.04 in.; water 2.31 ft.; kerosene 2.88 ft. By means of a suitable float, pen arm, and revolving clock, the U-tube

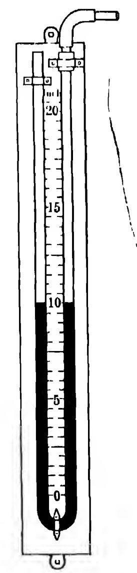


FIG. 645. Plain U-tube Manometer.

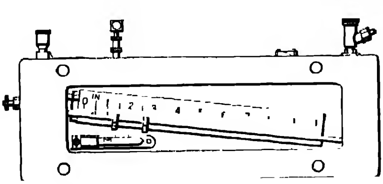


FIG. 646. Ellison Differential Draft Gage.

manometer may be designed for recording purposes as in Fig. 647. The differential readings may also be multiplied by means of suitable linkage, as for example, in the General Electric steam meter, Fig. 638. For very low pressure differences a compound liquid gage, such as the **Wahlen,** is sometimes used, but this is more of a laboratory than a power plant instrument.

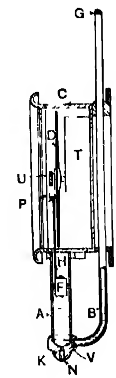


Fig. 647. Uehling Recording Vacuum Gage.

The **sylphon bellows** offers a sensitive and reliable means of indicating and recording small pressure differences, and dispenses entirely with the use of liquids. The "Precision" line of draft gages, Fig. 648, are based upon this principle and are constructed in single and multiple units, indicating and recording.

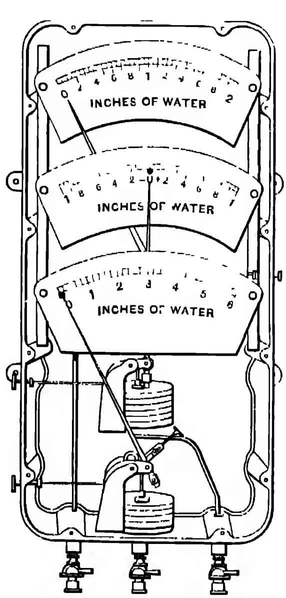


Fig. 648 Precision "3 in 1" Indicating Draft Gage.

Figure 649 illustrates the general principles of the Bailey recording draft gage, which is of the **bell-float** type. Two bells A and B are suspended from opposite ends of a beam (which is pivoted on knife-edge bearings) and are partly submerged in a light non-volatile oil as indicated. In measuring pressures less than atmospheric, connection is made at P_2, and P_1 is left open to the atmosphere. For pressures above atmospheric, connection is made at P_1, and P_2 is left open. For measuring the difference of two pressures, the higher pressure is applied at P_1 and the lower at P_2. If a slight suction pressure is applied at P_2, it is effective over the inside area of bell A and pulls it down into the liquid. The relative motion between bells A and B is transmitted through levers L and pen arm P to the recorder pen. This instrument may be designed to record pressures or pressure differences as low as 0.001 in. of water, though such low-pressure differential readings are subject to serious error because of the many influencing factors.

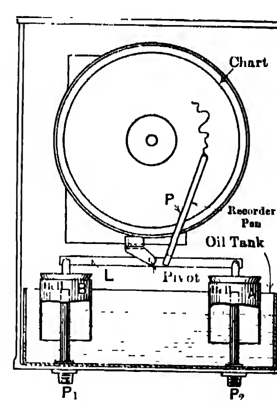


Fig. 649. Bailey Recording Draft Gage.

Measuring High Pressure with Dead Weight. Power, Feb. 26, 1916.

Combined Barometer and Vacuum Recorder: Power Plant Engrg., Feb. 15, 1923, p. 250.

344. Temperature Measurements. — The various types of thermometers and pyrometers which are available for measuring temperatures are outlined in Table 107. The temperature ranges given are the extremes for which the various types have been constructed, and while certain types are capable of measuring the entire range they are not necessarily suitable for all purposes.

The engraved or etched-stem **mercurial thermometer** is commonly used for special testing or laboratory purposes where the temperatures do not range beyond −35 to +900 deg. fahr. Only high-grade nitrogen-filled instruments should be used for temperatures over 400 deg. fahr. On account of their fragile construction and the difficulty of reading the scale, they are little used in power plants for permanent locations. The industrial type of glass thermometer, Fig. 650, while not as sensitive as the bare-bulb, is characterized by a heavy metal back and protecting tube for the bulb, large and distinct figures, and graduation marks, and threaded connections for attaching the instrument readily and firmly to some part of the apparatus. The etched-stem and industrial type of glass thermometers are indicating only, and must be placed close to the point where the temperature is to be taken. The errors in measuring temperatures with this class of thermometer are due, ordinarily, not so much to inaccuracies of the instrument as to the location and method of installation.

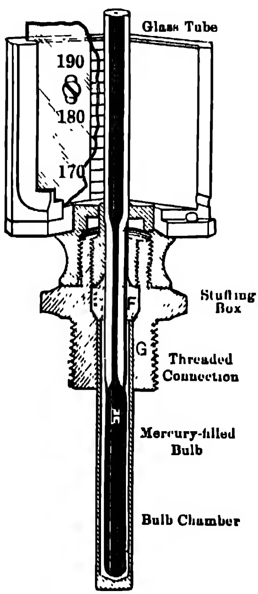


FIG. 650. Industrial Type of Mercury-in-glass Thermometer.

For indicating or recording purposes, the electrical, pressure, or bimetallic type of instrument is employed. Instruments utilizing the electrical or pressure principle permit of distant reading, but the bimetallic thermometer is the same as the mercurial glass instrument in this respect.

The **thermoelectric** thermometer or pyrometer has come into wide use as a reliable means of measuring temperatures from 100 to 2900 deg. fahr. It consists essentially of three parts: (*a*) the thermocouple of two different metals or alloys having a fused end (the hot junction), which is inserted where the temperature is to be taken, and the cold junctions which are opposite to the hot junction and which are maintained at some fixed temperature; (*b*) the indicator, which is either a millivoltmeter, a potentiometer, or a special type of instrument embodying both of these principles; and (*c*) two lead wires, usually of copper, connecting the cold

junctions of the thermocouple with the indicator. For obtaining continuous temperature-time curves, a recorder, operating on the same principles as the indicator, is used in place of the latter. In many cases, it is

TABLE 107

TYPES OF THERMOMETERS IN GENERAL USE

Principle of Operation		Type	Range in deg. fahr. for which they can be used
Expansion	Those depending on the change in volume or length of a body with temperature.	Gas	−400 to +2900
		Mercury, Jena glass, and nitrogen	−35 to +950
		Glass and petrol ether	−325 to +100
		Unequal expansion of metal rods	0 to 950
Transpiration and viscosity.	Those depending on the flow of gases through capillary tubes or small apertures	The Uehling	0 to 2900
Thermoelectric	Those depending on the electromotive force developed by the difference in temperature of two similar thermoelectric junctions opposed to one another.	Galvanometric	+100 to +2900
Electric resistance	Those utilizing the increase in electric resistance of a wire with temperature.	Direct reading on indicator or bridge and galvanometer	−330 to +1300
Radiation	Those depending on the heat radiated by hot bodies.	Thermocouple in focus of mirror	300 to 4000
		Bolometer.	−400 to Sun
Optical	Those utilizing the change in the brightness or in the wave length of the light emitted by an incandescent body.	Photometric comparison	1100 to Sun
		Incandescent filament in telescope	
		Nicol with quartz plate and analyzer.	
Calorimetric	Those depending on the specific heat of a body raised to a high temperature.	Platinum ball with water vessel.	32 to 3000
Fusion	Those depending on the unequal fusibility of various metals or earthenware blocks of varied composition.	Alloys of various fusibilities (Seger cones.)	32 to 3350

desirable to install both instruments as illustrated in Fig. 651. The thermocouples most frequently used are composed of platinum and platinum-rhodium (**rare-metal**) and chromel-constantan, copper-constantan, iron-constantan and chromel-alumel (**base metals**). The rare-metal

couples are suitable for temperatures up to 2900 deg. fahr., chromel-alumel up to 1800 deg. fahr., iron-constantan up to 1650 deg. fahr., and copper-constantan up to 1400 deg. fahr. When the "hot junction" of the thermocouple is heated, an electromotive force is set up which is a function of the temperature difference between its hot junction and its cold junction. The maximum e.m.f. developed by most base-metal couples, when operated at the highest safe working temperature, is somewhat less than 70 millivolts, and the platinum and platinum-rhodium couple develops an e.m.f. of about 16 millivolts.

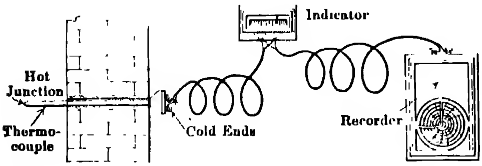


FIG. 651. Elements of a Typical Thermoelectric Pyrometer.

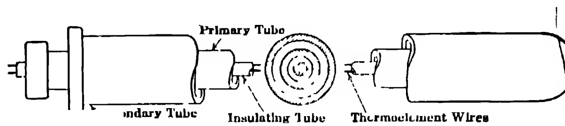


FIG. 652. Engelhard Rare-metal Thermocouple with Protecting Tubes.

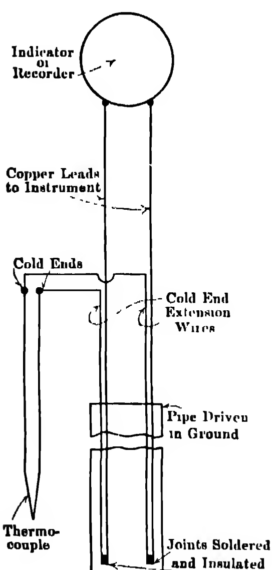


FIG. 653. Thermocouple with Cold-end Extension in Circuit and Cold Junction Buried in Ground.

In order to measure such small e.m.f.'s, a very sensitive galvanometer is required. Two types of instruments are used for this purpose, (1) the conventional **millivoltmeter** and (2) the **potentiometer.** Low-resistance millivoltmeters are more rugged and cheaper than the high-resistance instruments, but are subject to greater errors in case of change in circuit resistance. The latter are preferred where the leads are long and subject to considerable temperature variation. In either case, the temperature at the "cold junctions" must be kept constant where accuracy is essential, otherwise the readings will be in error. This is due to the fact that the e.m.f. developed by a thermocouple depends upon the temperature difference between its hot junction and cold junctions. Thus, for a constant hot-junction temperature, the e.m.f. will increase or decrease with decrease or increase in temperature of the cold junction. Corrections for variations in temperatures at the cold junctions may be made by use of compensating lead wires of practically the same material as the thermocouple,

terminating in a thermostatic cold-junction box or buried under-ground as shown in Fig. 653.

The most accurate method for measuring the e.m.f. of a thermocouple is by use of a potentiometer, the fundamental principle of which is illustrated in Fig. 654. A constant current from the battery B flows through the slide-wire resistance abc. One wire of the couple T is connected to the movable contact b and the other wire in series with a sensitive galvanometer is connected to a. The contact b is moved until the galvanometer reads zero, thus showing that no current is flowing through the thermocouple circuit. When this balance of zero setting is made, the true e.m.f. of the couple is equal to the potential drop across ab. The calibration of the scale is in no way dependent upon the constancy of magnets, springs, jewel bearings, level of the instrument, or variation due to ordinary changes in the resistance of the couple or of the lead wires. The entire potentiometer, galvanometer, battery, standard cell, slide wires, etc., as constructed, are mounted in a case not much larger than that of a milli-voltmeter. Indicating potentiometers are greater in cost than other types of instruments used for this purpose, and usually require manual adjustment for each setting. In the recorders the adjustment is automatic.

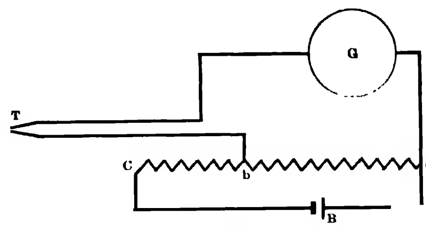


FIG. 654. Simple Wiring Diagram for Potentiometer Indicator.

Figure 655 shows a form of pressure thermometer which is used extensively for indicating and recording temperatures ranging from -60 to $+1000$ deg. fahr. It depends for its operation upon the pressure produced by a liquid or gas contained in a small bulb and exposed to the temperature to be measured. The pressure is transmitted to the indicating or recording mechanism through a flexible capillary tube. The indicating or recording element is ordinarily a pressure gage of the Bourdon-tube type, but diaphragm and liquid-manometer gages are also employed. The non-vaporizing liquid type is commercially limited to temperature ranges of -60 to $+200$ deg. fahr. with a length of connecting tubing not exceed-

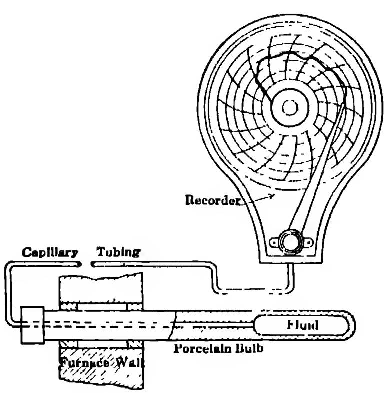


FIG. 655. Typical Pressure-type Thermometer with Recorder

ing 25 ft. The vaporizing liquid type is intended for temperatures ranging from +50 to +400 deg. fahr. with tubing lengths up to 300 ft. The gas-filled type is intended for temperatures ranging from +120 to +1000 deg. fahr. with tubing lengths up to 500 ft.

The resistance which most metals offer to the passage of an electric current through them varies with the temperature, a wire of given dimensions having a higher resistance when hot than when cold. By measuring the resistance of a coil of wire, an indication may be had of its temperature or that of the substance in which it is placed. The instruments used for measuring the resistance consist essentially of a Wheatstone bridge and a galvanometer. These instruments are either indicating or recording. **Electric-resistance** thermometers are suitable for accurately measuring temperatures from −330 to +1300 deg. fahr. and have the advantage over thermocouples in that any number of thermometer bulbs may be connected to an indicator by using a corresponding number of switches, and the scale of the instrument may be calibrated to cover any part of the total temperature range of the system. The distance between the thermometer bulbs and instruments may be as much as several hundred feet, provided there is little temperature variation in the leads.

For higher temperatures and for obtaining the temperatures of inclosed spaces above about 2500 deg. fahr., such as boiler furnaces, annealing ovens, and kilns, various forms of **optical** and **radiation pyrometers** have been devised. In such devices no part of the instrument is exposed to the temperature to be measured and hence the apparatus suffers no injury from this cause. Optical pyrometers are based upon the measurement of the brightness of the hot body by comparison with a standard.

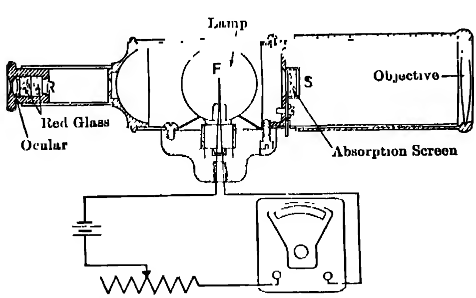


FIG. 656. Leeds and Northrup Optical Pyrometer.

The Leeds and Northrup optical pyrometer is shown in Fig. 656. The filament of a small electric lamp, *F*, is placed at the focal point of an objective *L* and "ocular", or eye piece. The assembly forms an ordinary telescope, which superposes upon the lamp the image of the sources viewed. Red glass is mounted at the ocular to produce approximately monochromatic light. In making a setting, current through the lamp is adjusted by means of a rheostat until the tip or some definite part of the filament is of the same brightness as the source viewed. The relation between current

through the lamp and temperature is either calculated or read from a table furnished by the manufacturers.

Other popular makes of optical pyrometers are the "Wanner," "Shore Pyroscope," "Scimatco," "F and F," and "Holborn-Kurlbaum."

Radiation pyrometers depend upon the measurement of the heat radiated from the hot body. The Féry radiation pyrometer, Fig. 657, is the best-known instrument of this type. When it is focused upon the source of heat, a cone of rays of definite angle is reflected by means of the mirror upon a thermocouple located in its focus. The electromotive force set up is measured in terms of the temperature of the source of heat by a millivoltmeter. Neither the couple nor any part of the instrument is ever subjected to a temperature much above 150 deg. fahr. The indications are practically independent of the distance from the source of heat, and the range is without limit. Other makes of radiation pyrometers are the "Thwing" and the "Foster."

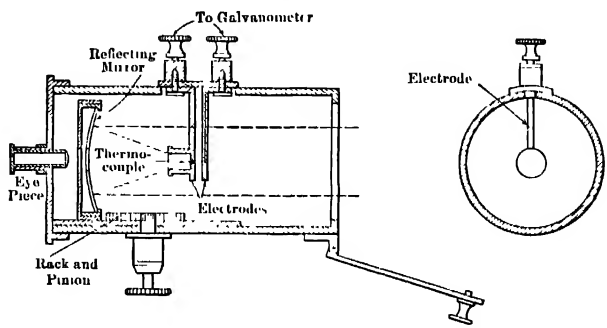


FIG. 657. Féry Radiation Pyrometer.

The **Uehling pyrometer** depends for its operation upon the flow of gas between two apertures, thus: Air is continuously drawn through two apertures by a constant suction produced by an aspirator. So long as the air has the same temperature in passing through these orifices, there is no change in the partial vacuum in the chamber between them; if, however, the air passing through the first opening has a higher temperature than that passing through the second, the vacuum in the chamber will increase in proportion to the difference in temperature since the volume of air varies directly with the temperature. In the application of this principle, the first aperture is located in a nickel tube which is exposed to the heat to be measured, while the second aperture is kept at a uniform lower temperature. This style of pyrometer is made to indicate and record, and the indicating and recording mechanism can be placed at a distance from the main instrument.

Bimetallic thermometers utilizing the turning moment produced by the differential expansion of two metals brazed together, or the linear differential expansion of two rods having different coefficients of expansion are used for certain industrial processes where the temperatures range from

-40 to $+500$ deg. fahr., but are not much in evidence in the power house. The bimetallic principle, however, is used in a number of automatic temperature-controlling systems.

Pyrometric Practice: U. S. Bureau of Standards, Technologic Paper No. 170, Feb. 16, 1921.

345. Power Measurements. — Instruments for the measurement of power may be divided into two general classes, **direct** and **indirect.** The former involve the direct measurement of force and linear velocity or torque and angular velocity, and the latter give the equivalent in other forms of energy. Direct power-measuring appliances include the various speed indicators, transmission and absorption dynamometers; and the indirect include ammeters, voltmeters, watt-hour meters, boiler-flow meters, and the like. In all power measurements the time or speed factor is readily determined, but the force or torque factor, or equivalent, often involves considerable labor and the use of costly and complicated apparatus. The various conversion factors for the measurement of work, power, and duty are given in Appendix A.

346. Measurement of Speed. — The following chart gives a classification of a number of well-known instruments for determining linear and angular velocities.

Counters	Hand	Worm and wheel.
	Continuous	Gear train. Electrical.
Tachometers or Speed Indicators	Centrifugal	Weights. Liquids.
	Electrical	
	Resonance	Frahm's.
Chronographs	Electromagnetic Tuning fork.	

The most commonly used device for speed determinations is the **hand speed counter,** consisting of a worm, worm wheel, and indicating dials. The errors to be corrected are principally those due to slipping of the point on the shaft, and to the slip of the gears in the counting device in putting in and out of operation. In some of the better grades of instruments, the gears are engaged or disengaged with the point in contact with the shaft. In the latter design a stop watch, actuated by the disengagement gear, minimizes the error likely to occur in hand manipulation.

The **continuous counter** consists of a series of gears arranged to operate a set of indicating dials. It may be operated by either rotary or reciprocating motion. The rate of rotation is calculated from the readings of the counter.

All **tachometers** indicate directly the speed of the machine to which they are attached and are independent of time determination. The most

commonly used devices depend upon the centrifugal force of revolving weights for their operation. The indicating needle is attached to the weights in such a manner that the number of revolutions per minute is read directly from the position of the needle on the dial. These instruments should be calibrated for accurate work because of the number of wearing parts.

Fluid tachometers consist essentially of small centrifugal pumps or blowers discharging into a suitable type of manometer. The height of the indicating column is a function of the speed of rotation. The application of this type of tachometer is found in the Bailey recording stoker tachometer. In this particular design the suction side of a small centrifugal blower is attached to one bell and the discharge to the other bell of a Bailey draft recorder.

Electrical tachometers are miniature dynamos, the voltage being a measure of the speed of rotation. These instruments are accurate and readily attached but necessitate the use of a delicate and costly voltmeter. The indicating mechanism may be placed at any distance from the small dynamo and in this respect has a marked advantage over the other types of speed indicators.

The **resonance tachometer** affords a convenient method of measuring speeds over a wide range. It consists of a number of steel reeds of different periodicity mounted side by side on a suitable frame. When used to measure the speed of an engine or turbine, the instrument is placed on or near the bed plate or frame and the slight under or over balance causes the proper reed to vibrate in unison.

347. Steam-engine Indicators. — This subject has been extensively treated by various authorities and a general discussion would be without purpose. For indicated horsepower, testing indicator springs, and analysis or indicator diagrams see "Rules for Conducting Steam Engine Tests," A.S.M.E. Code of 1925.

348. Dynamometers. — Dynamometers for measuring power are of two distinct types, **absorption** and **transmission.** In the former the power is absorbed or converted into energy of another form, while in the latter the power is transmitted through the apparatus without loss, except for minor friction losses in the mechanism itself.

The ordinary **Prony brake** is the most common form of absorption dynamometer. In the various forms of Prony brakes, the power is absorbed by a friction brake applied to the rim of a pulley. For low rubbing speeds and comparatively small powers it affords a simple and inexpensive means of measuring the actual output.

The Alden absorption dynamometer is a successful form of friction brake and has a wide field of application. It has been constructed in

large sizes and is adapted to all practical ranges of speed. For a description of rope brakes and the Alden absorption dynamometers see Appendix No. 19, p. 179, A.S.M.E. Code of 1915.

Water brakes are finding much favor with engineers for high-speed service. There are two types, the Westinghouse and the Stumpf. In the former, the rotor consists of a simple drum with serrated periphery, revolving in a simple casing, the inner surface of which is serrated in a manner similar to the rotor. The resistance is produced by friction and impact, and the power is converted into heat which is carried away by the circulating water. The casing is free to turn about the shaft but is held against rotation by a lever arm. The torque of the lever arm is determined as in a Prony brake. A brake of this design, 2 ft. in diameter and 10 in. wide, will absorb about 3000 hp. at 3500 r.p.m. In the Stumpf type, the rotor consists of a number of smooth discs mounted side by side on a common shaft. The casing is divided into a number of compartments corresponding to the division of the rotor. There is no contact between rotor and casing. The friction between the discs and water and the water and casing tends to rotate the latter and the torque is measured in the usual way. In either type, the power output is readily controlled by the water supply.

Pump brakes and **fan brakes** are also used as absorption dynamometers. The latter are commonly used in connection with automobile engine testing.

Electromagnetic brakes are occasionally used for power measurements. They consist essentially of a metal disc or wheel revolving in a magnetic field. The resistance or drag tends to revolve the field casing and the torque is measured in the usual way.

An electric generator mounted on knife edges forms the basis of the Sprague electric dynamometer. The prime mover drives the armature of the generator and the reaction between armature and field is counterbalanced by suitable weights. The output is conveniently regulated by a water rheostat.

Transmission dynamometers are seldom used for testing prime movers and are ordinarily limited to small power measurements. In some instances, however, as in marine service, transmission dynamometers afford the only practical means of approximating the net power delivered to the propeller. For comparatively small power measurements may be mentioned the Morin, Kennerson, Durand, Lewis, Webber, and Emerson transmission dynamometers, and for large powers, the Denny and Johnson electrical torsion meter and the Hopkinson optical torsion meter. For detailed descriptions of these appliances consult "Experimental Engineering," Carpenter and Diederichs, Chap. X.

349. Flue-Gas Analysis. — It has been shown (paragraph 46) that the products of combustion, commonly called flue gases, resulting from the complete oxidation of coal with theoretical air supply, consist chiefly of nitrogen and carbon dioxide, with lesser amounts of water vapor and sulphur dioxide. It was also shown that with incomplete combustion the flue gases may contain carbon monoxide and varying amounts of hydrocarbon. If excess air were used in the combustion of the fuel, free oxygen would also be present in the gases. Evidently an analysis of the flue gases offers a basis for judging the efficiency of combustion. The first step in the analysis, and the most important one, is the obtaining of a representative sample. Since the gases in the breeching and flues may be far from homogeneous, great care must be exercised in getting a true average sample. (Sampling and Analyzing Flue Gases, U. S. Bureau of Mines, Bul. No. 97, 1915.)

The analysis as ordinarily made in commercial practice is called volumetric, although in reality it is based upon the determination of partial pressures. According to Dalton's laws, when a number of gases are confined in a given space each gas occupies the total volume at its own partial pressure, and the total pressure is the sum of all the partial pressures. When one of the gases is absorbed by a suitable medium and the remaining gases are compressed back to the original total pressure, a volume decrease is found, and if the temperature remains constant this decrease represents the volume absorbed.

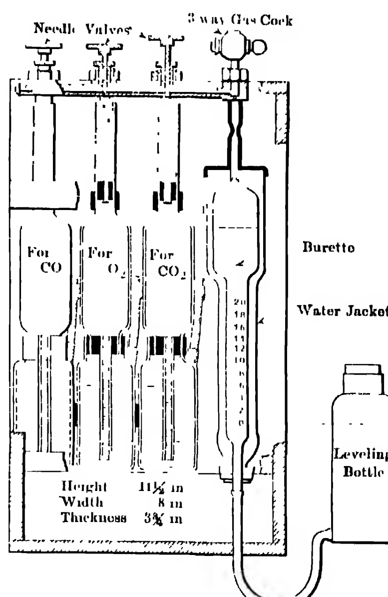


Fig. 658. Orsat Apparatus (Hays Improved Gas Analyzer).

The apparatus usually employed for volumetric analysis consists of a graduated measuring tube into which the gases are drawn and accurately measured under a given pressure, and a series of treating tubes, containing the necessary absorbing reagents, into which they are transferred until absorption is complete. The **Orsat apparatus,** Fig. 658, forms the basis of nearly all of the portable appliances on the market for analyzing flue gases and the ordinary products of combustion. In this apparatus a measured volume, representing an average sample of the gas, is forced successively through

pipettes containing solutions of caustic potash, pyrogallic acid, and cuprous chloride in hydrochloric acid, respectively, thus absorbing the carbon dioxide, the oxygen, and the carbon monoxide, the contraction of volume being measured in each case. The exact process of measuring the gas sample, its transference to the pipettes and manipulation of the various valves differ with each design of apparatus and the instructions of the manufacturer should be faithfully followed. For a comprehensive discussion of analyses of flue gases in general, consult "Analysis of Flue Gases" by Henry Kreisinger and F. K. Ovitz, Bureau of Mines, Bul. No. 97, 1915.

The **Hempel apparatus** works on the same principle as the simple form of Orsat apparatus described, so far as the latter is applicable, excepting that the absorption may be hastened by shaking the pipettes bodily, bringing the chemical into intimate contact with the gas. The Hempel apparatus is less portable and requires more careful manipulation than the Orsat, and for this reason is more of a laboratory than a power plant instrument. The absorption pipettes are made in sets, which are in the form of globes, and a number of independent sets are required for the treatment of the different constituent gases. A simple pipette of the Hempel type is shown in Fig. 658*a*.

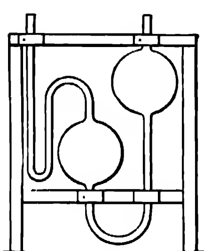


FIG. 658a. Hempel Pipette.

The standard Orsat apparatus is equipped for the analaysis of CO_2, CO, and O_2 only, but some designs are on the market in which provision has been made for the analysis of illuminants, hydrogen, and methane in addition to these three gases.

For rough surveys or where there are but small amounts of CO, H, and hydrocarbons, no attempt is made to analyze other than the CO_2. A number of small inexpensive portable devices are available for this purpose, among which may be mentioned the "Dwight," "Republic Flue Gas Analyzer," "Hays Single Pipette Gas Analyzer," and "Bacharach Pocket CO_2 Indicator."

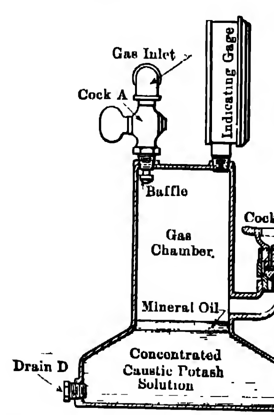


FIG. 659. Dwight CO_2 Indicator.

The general principle of the **Dwight CO_2 Indicator** is illustrated in Fig. 659. It consists essentially of a small hard-rubber vessel equipped with a sensitive vacuum gage and partly filled with a solution of caustic potash. A film of oil seals the potash solution against contact with air or

gas when the apparatus is not in use. Operation is as follows: A sample of gas is introduced into the vessel by means of a rubber pump. The inlet valve and the valve connecting the gage with the vessel are closed. The vessel is then shaken violently so that the oil film is broken and the gas and caustic thoroughly mixed. The volume of gas absorbed is indicated by the vacuum registered on the gage.

The majority of CO_2 indicator recorders are of the absorption type; that is, the measurements are controlled by the absorption of CO_2 by KOH, either liquid or granular. Among the well-known instruments of this type may be mentioned the "Republic," "Hays," "Precision," "Foxboro," "Tag-Mona," and "Uehling." The motive power for actuating the mechanism may be steam, water, flue draft or electricity.

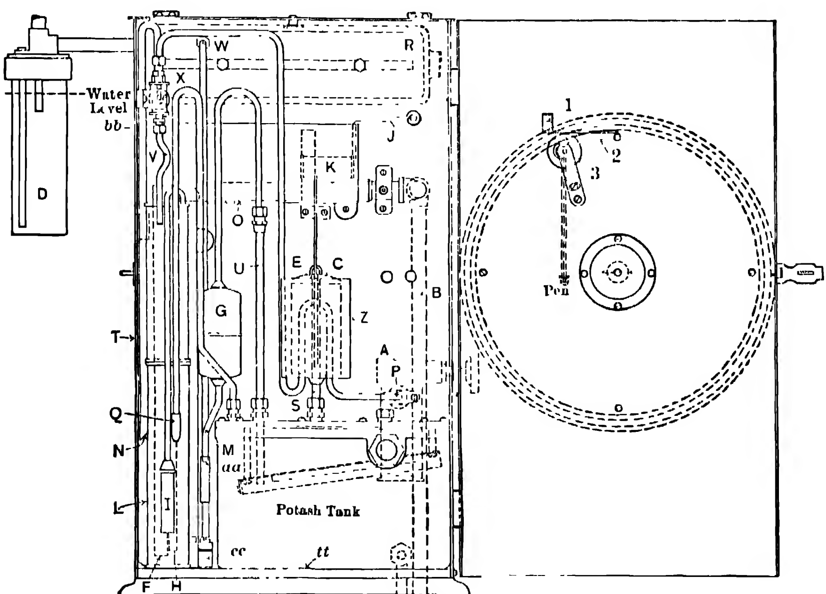


FIG. 660. "Republic" CO_2 Recorder (Front Door Open).

Figure 660 shows a general assembly of the "**Republic**" **CO_2 Recorder,** which is representative of the absorption type. This instrument depends upon the flow of water for its operation, but is so constructed that a considerable variation in supply will not affect its proper functioning. The gas is drawn continuously through a suitable filter from the furnace or breeching to the instrument by means of an aspirator of the water-jet type.

When the CO_2 recorder is in operation, the water admitted through

strainer and valve *A*, Fig. 660, enters the instrument at connection *P*, and flows through the loop in the oil seal *Z* for the purpose of maintaining constant temperature of the oil. It then passes vertically to the top of the case and follows through the jet X and aspirator *V* into water column *T*. The water flowing through the jet forms a partial vacuum, drawing the gas from the furnace through the indicator bottle *D* following the arrows to the aspirator.

The water rising in the water column also rises in the vertical tubing connected to the water column at *CC*. When the water reaches the tubes *aa*, it seals off a definite volume of gas contained in the pipette *G*. As the water continues to rise, it fills the pipette, forcing the contents of the pipette through the gas tube *U* into tank *tt* containing the potash solution. The gas passes through the potash solution under the entire length of the baffle plate in the tank and rises to the surface. This action absorbs all the CO_2 contained in the gas, leaving an amount of residue gas which escapes through outlet *S* and raises float *C* to a height proportional to the amount of residue gas.

As the water continues to rise in the water column, it raises the siphon system until it strikes bumper *bb*. The water then overflows into the float of siphon *L*, causing the siphon system to sink. When the siphon is submerged, the water is siphoned out of the water column through tubes *F*, *H* and *O* connecting with drain pipe *B*. As the water level is lowered below *Q*, it releases the seal of the tube leading to the waste gases through the potash tank. The float *C* lowers to rest on guide *E* and discharges the contents of gas through relief tube *Q*.

As the receding water reaches point *aa*, the pipette discharges through pipe *M*, drawing a charge of gas from the gas passage at *W*. When the water recedes to point *H*, air is admitted and the siphon action stops. At this time the water siphons from the main siphon *L* through the auxiliary siphon *I*.

When the pipette is measuring the charge of gas, the partial vacuum in it and in the connecting gas chambers is relieved by vent tube *N* which supplies air to satisfy the requirements of that partial vacuum and the aspirator. The water column begins to fill again, repeating the operation.

The recording mechanism is mounted on the door. When the door is closed, arm 2, rigidly connected to recording pen staff, swings into position directly over rod attached to float *C*, and pen retainer 1 is directly over float arm at *J*. The final movement of the water raises float *K* and releases friction surface of 1 from wheel 3. This allows pen arm 2 to rest freely on the float rod, registering the amount of CO_2. As the water recedes, pen retainer 1 rests on the wheel holding the pen in position

until the next charge. This gives a continuous chart record, and the added feature of an indicating instrument.

The **Uehling Composimeter** is another successful instrument for continuously recording the percentage of CO_2 in the flue gas. The principles of this apparatus are illustrated in Fig. 661. The device consists primarily of a filter, absorption chamber, two orifices, *A* and *B*, and a small steam aspirator. Gas is drawn from the usual source, by means of the aspirator, through a preliminary filter located at the boiler, and then through a second filter as illustrated in the diagram. From the latter the gas passes through orifice *A*, thence through the absorption chamber and orifice *B* to the aspirator where it is discharged. The CO_2 is absorbed by the caustic potash solution in the absorption chamber. This reduces the volume and causes a change in tension between the two orifices in proportion to the CO_2 content of the gas. This variation in tension is indicated by the water column, as shown, and is transmitted by suitable piping to the recording mechanism which may be placed at a considerable distance from the boiler room.

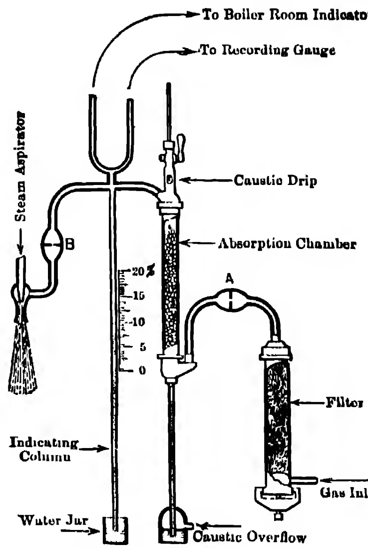


FIG. 661 Principle of the Uehling CO_2 Recorder.

Figure 662 gives a diagrammatic outline of the **Engelhard Gas Analyzer** which operates on the thermal conductivity basis and does away entirely with liquids and gas-absorbing reagents. The fundamental principle is that of comparison of two resistance wires electrically heated, one surrounded by a standard or reference gas, the second by the gas being analyzed. These wires are connected into a Wheatstone bridge system as shown. Each of the two resistance wires, which are of 0.6 mm. diam-

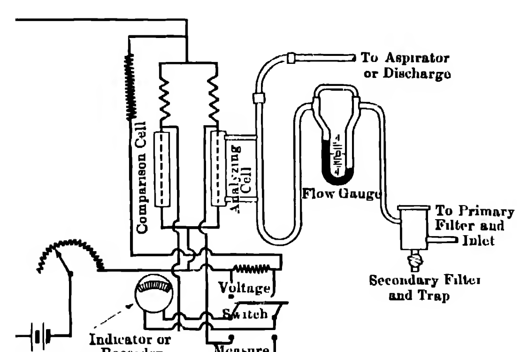


FIG. 662. Principle of Engelhard Gas Analyzer.

eter platinum, are mounted in a heavy-wall copper tube of small inside diameter. When the two gases are identical in composition, both wires are at the same temperature and the bridge is in balance. If the thermal conduction of one gas is different from that of the other, there will be a temperature difference in the wires and this will effect an unbalanced condition of the bridge. This unbalanced effect may be indicated and recorded by suitable instruments calibrated to read directly in terms of the gas being analyzed. Electrical gas analyzers have so many advantages over the caustic or absorption type that it is safe to predict that they will eventually supersede the latter for continuous service, indicating or recording.

350. Combination Instruments. — Instead of employing an individual dial or chart for each indicating or recording instrument, the various indications or graphs may be grouped in a single instrument. This not only makes it possible to visualize the simultaneous readings of pressures, temperatures, and the like, but also offers a means of increasing the efficiency of operation without a knowledge of the actual values of the quantities involved. For example, the heat loss in the dry chimney gas is a product of weight, mean specific heat, temperature of the air entering the furnace, and temperature of the flue gas. Since the mean specific heat is practically constant for the temperature range in practice, and that of the air entering the furnace varies within a comparatively narrow range, it is evident that the heat loss is primarily dependent upon the product of the weight and temperature of the flue gas. It has been shown that for a given class of fuel the per cent of CO_2 in the flue gas is an index of the weight of gas. Therefore, a single chart upon which the variation in CO_2 and flue-gas temperature is recorded is substantially a relative-efficiency meter. Thus any change in the method of firing or operation which lowers the temperature reading and at the same time increases the CO_2 content (within the maximum per cent of CO_2 permissible for the particular installation under consideration) will result in decreased stack losses irrespective of the actual temperature and CO_2 content.

The rate of flow of the flue gas for a given grade of fuel and a given boiler equipment is a function of the draft-pressure drop between fire box and uptake or between passes in the boiler, because the resistance of the gas passages may be likened to an orifice. Therefore, a simple chart upon which the variation in draft-pressure drop and flue-gas temperature is recorded performs duties similar to those of the combination instrument previously described.

The weight of air required for the complete combustion per lb. of a given grade of fuel is a definite amount, and the heat generated per lb.

of fuel is equally definite; therefore, for a boiler and furnace equipment, grade of fuel, and load, there is a definite relationship between steam output and air flow. An instrument giving combined readings of steam flow and air flow is therefore of value in maintaining efficient combustion at various ratings.

In a similar manner, various readings may be combined on one chart. Accumulation of ash and soot on the tubes, leaky settings, and broken baffles will, of course, influence air flow readings based on pressure drops, and incomplete combustion may greatly offset the value of high CO_2 readings; but, taking all things into consideration, these various types of combination or relative efficiency instruments have considerable merit and may be the means of effecting increased economy if properly installed and intelligently studied.

Among the well-known instruments may be mentioned the following:

Bailey Boiler Meter, combining, on a simple graphical chart readings of steam flow, air flow, and flue-gas temperatures, and in special cases, temperature of the ash leaving the grate. Fire-box draft indications may also be included.

Republic Steam CO_2 Recorder, giving continuous records of the steam flow and CO_2 content.

Hays Automatic CO_2 *and Draft Recorder.*

Engelhard Combination CO_2 *and Flue-Gas Indicator-Recorder.*

351. Boiler Control Boards.—In the large modern central station, efficient operation of the various units composing the plant is greatly

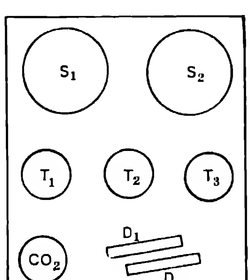


FIG. 663. Individual Boiler Control Board.

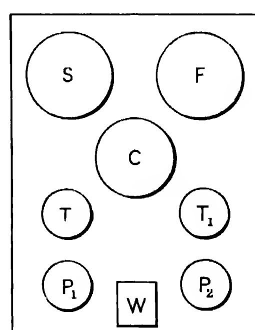


FIG. 664. Boiler Section Control Board.

facilitated by grouping the testing instruments on a control board and by placing this board where it can be conveniently studied by the operating engineer. Figure 663 shows the individual control board as installed before each boiler unit in the Northwest plant of the Commonwealth

Edison Company of Chicago, and Fig. 664 shows the section control board for each turbine unit. The individual control board is mounted on the front of the boiler casing, and the section board is placed at the end of the battery of boilers near the wall dividing the boiler from the turbine room. With reference to Fig. 663, the two instruments at the top are steam-flow meters — one on each steam lead — with indicating, recording, and integrating attachments. These meters show the amount of steam delivered at any time by the boiler and give a complete record of its delivery. The three recording gages below show the temperature in uptake from the boiler, the temperature of the feedwater leaving the economizer and entering the boiler, and the temperature of the flue gases leaving the economizers. Below and at the left is a CO_2 recorder, while at the right-hand corner are two indicating draft gages, one connected to the furnace and the other to the uptake. With reference to the section control board, the two flow meters at the top measure the steam input to the turbine and the feedwater input to the boilers, respectively. The recording thermometers immediately below show the temperature of the steam entering the turbine and the temperature of the feedwater entering the economizer, respectively. Below these are two recording pressure gages showing the pressure on the steam header and on the boiler feed header, respectively, while in the center of the board is a clock and below that an indicating wattmeter showing the output of the turbo-generator unit which is direct connected to these boilers. Where automatic coal-weighing devices are in use, the individual control board includes the fuel-measuring dials. By the use of these instruments a very complete check is obtained of the performance of individual boilers of the entire unit.

352. Steam Calorimeters. — Several forms of calorimeters are available for determining the quality of the steam. The simplest, as well as the most satisfactory, if the percentage of entrained moisture is not beyond its range, is the **throttling** calorimeter, Fig. 665. In this device the sample of steam, which is taken from the steam pipe by means of the perforated nipple, is allowed to expand through a very small orifice into a chamber open to the atmosphere. The excess of heat liberated serves first to evaporate any moisture present and then to superheat the steam at the lower pressure. From the observed temperatures and pressures, it is easy to calculate, with the aid of steam tables, the percentage of moisture in the original sample. See paragraph 391.

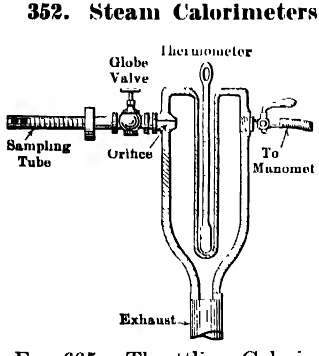


Fig. 665. Throttling Calorimeter.

The limit of the throttling calorimeter depends upon the steam pressure and is about 3 per cent of moisture at 80 lb. pressure and about 5 per cent at 200 lb. For steam containing greater percentages of moisture, the **separating** calorimeter, Fig. 666, is sometimes used. This instrument is virtually a steam separator and mechanically separates the moisture from the sample of steam. The water thus separated collects in a reservoir provided with a gage glass and a graduated scale, while the dry steam passes through an orifice to the atmosphere. The weight of dry steam per unit of time is indicated on the gage, calculated according to Napier's rule, or may be determined by condensing and weighing. The accuracy of the moisture determination is greatly affected by the difficulty of obtaining true samples of steam containing large percentages of moisture.

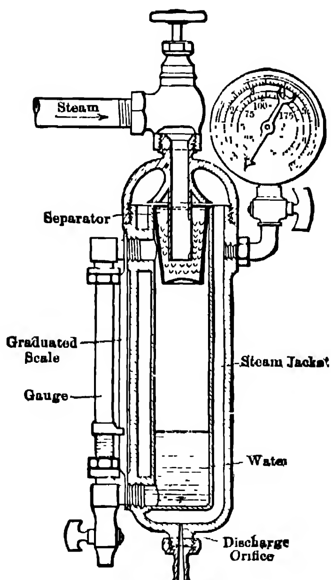


FIG. 666. Separating Calorimeter.

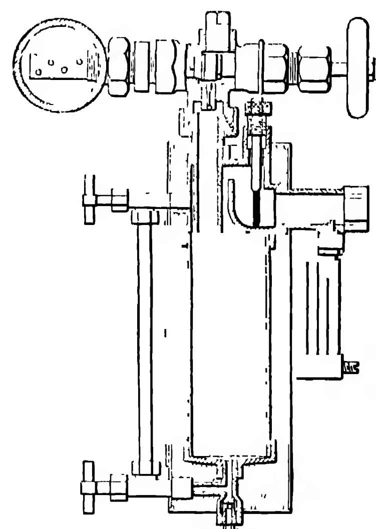


FIG. 667. Universal Calorimeter.

Figure 667 shows the Ellison **universal** steam calorimeter, which combines the separating and throttling principles and is adapted to steam of any degree of wetness. The separating chamber is provided with a gage glass, not shown, for indicating the weight of water which accumulates only when the steam is too wet to be superheated.

Throttling Calorimeters: Power, Dec., 1907, p. 891; Trans, A.S.M.E., 17–151; 175, 16–448; Engr. U. S., Feb. 15, 1907, p 219

Separating Calorimeters: Trans. A.S.M E., 17-608; Engr. U. S., Feb. 15, 1907, p. 219.

Universal Calorimeter: Trans. A.S.M.E., 11–790.

Thomas Electrical Calorimeter: Power, Nov., 1907, p. 791.

353. Fuel Calorimeters. — The analysis and heat evaluation of fuel require considerable time and skill and much costly apparatus; hence in most power plants it is customary to depend upon a specialist to whom samples are submitted from time to time. In many large stations, however, the conditions often warrant the establishment of a testing laboratory equipped for the proximate analysis of coal and the determination of the calorific value of the solid, liquid or gaseous fuel used. Calorimeters of the **Mahler bomb** type, Fig. 668, are the most accurate and satisfactory devices for solid and liquid fuels, but are comparatively expensive. These instruments consist of a steel shell or "bomb" of great strength, lined with porcelain or platinum, into which a weighed sample of the fuel is

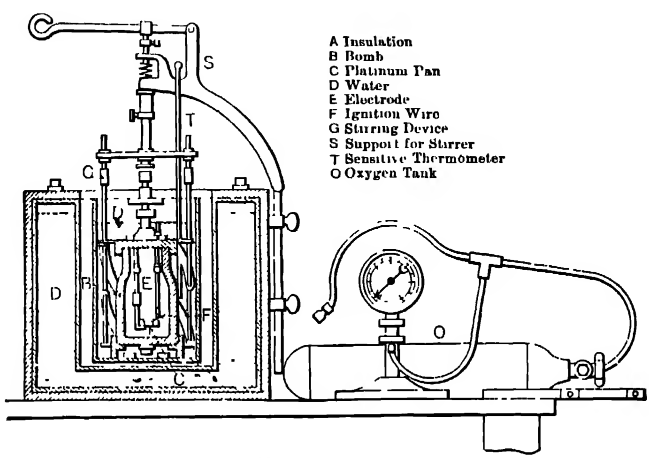


Fig. 668. Mahler Bomb Calorimeter.

introduced and burned on a platinum pan in the presence of oxygen under a pressure of about 300 lb. per sq. in. The charge is ignited by an electric current. During combustion the bomb is submerged in a known weight of water, which is kept constantly agitated. The calorific value is calculated from the observed rise in temperature due to the heat evolved, proper corrections being made for the water equivalent of bomb and appurtenances, for the heat given up by the igniting current, and for radiation or absorption of heat from the surrounding air.

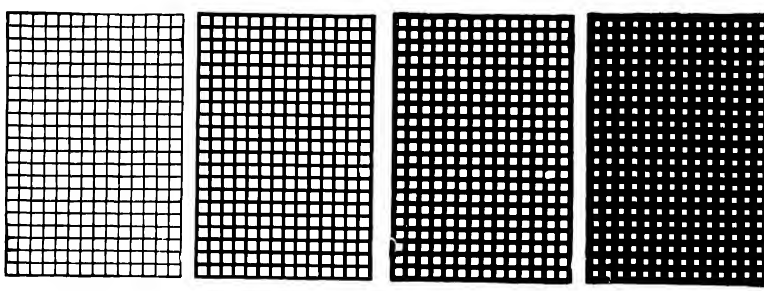
In the "adiabatic" design, radiation correction is made unnecessary by surrounding the inner water vessel with a water jacket, the temperature of which is automatically maintained the same as that in the inner vessel. In some of the very latest designs, the inner water vessel is insulated by a vacuum jacket similar to a thermos bottle.

The heat value of gaseous fuels is obtained by calorimeters of the

"**Junker**" type, which are essentially small tubular gas heaters in which a very small temperature difference is maintained between the inlet and outlet water and the flue gas escapes at a temperature which is essentially that of the gas and air supply.

354. Smoke Determination. — Smoke measurements may be either quantitative or relative.

The most satisfactory method, at this writing, of determining the quantity of smoke passing through a chimney is that adopted by the Chicago Association of Commerce. A continuous sample of chimney gas is drawn from the stack by means of a special Pitot tube and exhauster, and the solid particles are entrapped in a filter. The tube is so arranged that the rate of flow through the apparatus is the same as that in the chimney. Since the area of the tube opening bears a fixed ratio to that of the chimney, the weight of carbon, cinders, soot, and the like caught in the tube filter is a measure of the total weight emitted from the stack.



No. 1. No. 2. No. 3. No. 4

FIG. 669. Ringelmann Smoke Chart (Greatly Reduced).

Quantitative measurements are of considerable value in estimating the amount of energy lost in the production of visible smoke, but are seldom attempted in regular practice.

There are several methods of determining smoke, relatively. The most common is that devised by Ringelmann, and is commercially known as the **Ringelmann Smoke Chart**. The chart, as commonly used, consists of a cardboard folder 12 by 26 in. over all. Four charts are printed on this folder, each chart consisting of 294 squares, 14 squares wide by 21 squares in length, the width of the lines and spacings varying as follows:

No. of Card	Thickness of Lines, Mm	Distance in the Clear between Lines, Mm.
1	1	9.0
2	2.3	7.7
3	3.7	6.3
4	5.5	4.5

At a distance of 50 ft. from the observer, the lines become invisible and the cards appear to be of different shades of gray, ranging from very light gray almost to black. The observer places the chart on a level with the eye (at the distance stated, and as nearly as possible in line with the chimney) and notes which card most nearly corresponds with the color of the smoke. Observations should be made at 15-second intervals and recorded as in Fig. 670. No smoke is recorded as No. 0, 100 per cent as No. 5, and the intermediate colors as indicated by the cards.

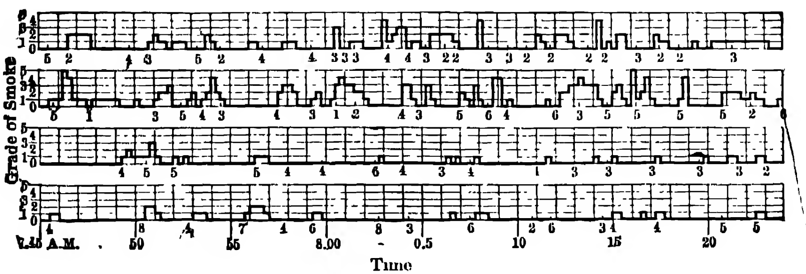


FIG. 670. Smoke Record Chart.

Experienced observers often record in half-chart numbers. Although these observations depend upon the personal element, it is the opinion of the Chicago Smoke Department that only a little experience is necessary to effect consistent results with different observers. Observations are made on a given stack every 15 seconds throughout the entire day and the total "smoke units" are recorded, from which the average smoke density for the entire period is calculated.

A "smoke unit" is the equivalent of No. 1 smoke (Ringelmann scale) emitted for one minute. No. 1 smoke has a density of 20 per cent; No. 2, 40; No. 3, 60; No. 4, 80; and No. 5, 100 per cent. Thus, if a stack emits No. 3 smoke for 6 minutes, 18 smoke units are charged against it. If this smoke was emitted during one hour's observation, then

$$3 \times 6 \times 20/60 = 6 \text{ per cent}$$

is the average density of smoke emitted during the period of observation.

Smoke recorders which project a continuous stream of the chimney gases against a clock-operated chart, and in this manner automatically record the density of the smoke, are on the market but have found little favor with engineers because of high first cost.

One of the most successful instruments for showing the density of the smoke, and one which may be placed so that it is plainly visible to the fireman, is the **Eclipse Smoke Indicator.** This device may be likened to a periscope with one end connected to the stack or breeching and the

other placed at a convenient point in the firing aisle. An incandescent lamp with reflector is placed in the stack or breeching directly opposite the periscope opening and projects a beam of light across the stack or breeching. This beam of light is transmitted through the tube to the glass indicating dial in the boiler room. The intensity of the light is affected by the amount of visible smoke in the escaping gases and the variations are instantly shown on the indicating dial.

CHAPTER XIX

FINANCE AND ECONOMICS. — COST OF POWER

355. General Records. — In many states, public utility corporations are required to submit an annual statement covering the various details of operation, and, in order to insure uniformity, ruled and printed forms are furnished by the state. The private plant owner, on the other hand, is free to use his own judgment and may adopt any system of cost accounting or dispense with it entirely. In all power plants, public or private, an itemized record of plant performance and cost of operation is of vital importance for the most economic results.

The principal objects of keeping a system of records are (1) to enable the owner to accurately determine the power plant operating cost, and (2) to enable the operator to analyze the various records with a view of reducing all losses to a minimum. Power-plant records, to be of value, must be closely studied with a view toward improvements. The mere accumulation of data to be filed away and never again referred to is a waste of time and money.

Records should cover not only the daily, monthly, and yearly operation of the plant but also, as permanent statistics, a complete analysis of each item of equipment. The value of such data cannot be overestimated. The engineer will frequently find it greatly to his interest to have available the complete details of the renewable parts of the equipment when it is required to replace a broken or worn-out part in case of emergency.

A number of attempts have been made to standardize power-plant records but the results have been far from satisfactory because of the wide range in operating conditions. Each installation is a problem in itself and the items to be recorded must necessarily depend upon the size and character of the plant. A common mistake is to attempt too comprehensive a system, with the result that after the novelty has ceased the labor of making the various entries becomes irksome, many of the items are omitted, guesses are substituted in place of actual observations, and the records are ultimately without value. A few properly selected items, accurately recorded, are of vastly more importance than an elaborate system of records indifferently maintained.

Walter N. Polakov, *Trans. A.S.M.E.*, Vol. 38, 1916, p. 581, has proposed a " standardization of power-plant operating cost " by means of

which the owners of power plants can judge, without the necessity of going into technical details themselves, how closely the actual performance of the plant is to the possible minimum cost at any time or under any circumstances, all variable factors beyond operating control being automatically adjusted. Mr. Polakov shows the futility of attempting to judge any one plant by the performance of others having a different kind of equipment or of a different nature of service. Even where conditions appear identical, such comparisons do not offer a true measure of excellence. It is not so important to know that one's plant is better than another as to know whether it is as good as it can be. Mr. Polakov shows how this can be determined by the use of curves of "standard costs," the plotting and application of which are explained in his paper before the American Society of Mechanical Engineers.

TABLE 108

PERMANENT STATISTICS

General Information

Date of installation		Ground plan	191 × 231
Type of building	Office	Rentable floor space, sq. ft.	400,000
Number of floors	18	Height of building, ft.	280
Number of offices	900	No. of sides exposed	3
Volume of building, cu. ft.	10,860,000	Radiator surface, sq. ft.	100,000
Type of heating system	Webster	Glass surface, sq. ft.	100,000
Engine room, sq. ft	6,840	Boiler room, sq. ft.	5,400
Height of chimney, ft.	318	Number of elevators	22
Draft, in. of water	3 5	Type of elevators	High pressure hydraulic
Kind of grate or stoker	Jones Underfeed	Capacity of elevators, lb., each	2,700
Kind of coal	Ill. screenings	Boiler pressure	150
Coal storage capacity, tons	450	Back pressure	Atmospheric
Capacity ice plant, tons	50	Part of bldg. lighted	All
Capacity storage battery, am. hr.	None	Total cost of mechanical plant	$650,000
Total cost of building	$5,000,000		

	Engines	Generators	Motors	Boilers
Type	Ball compd.	Crocker-Wheeler		
Number installed	5	5	25	5
Rated capacity	250 hp.	150 kw.		375 hp.

R. J. S. Pigott, *Trans. A.S.M.E.*, Vol. 38, 1916, p. 687, shows, by means of graphic analysis, the effects of modifying the operating conditions of power plants and of changing the character of the auxiliary equipment. From the study of such an analysis the cost of producing power for given conditions may be determined with little effort, and the effects

of changes in the conditions or equipment may be predetermined with accuracy.

The National Association of Building Owners and Managers have recommended a standard form of statement, outlining the classification of accounts for office buildings, which may be obtained from their Executive Office, Edison Bldg., Chicago, Ill. This association also issues an "Experience Exchange" which gives the cost of operating the mechanical equipment of office buildings in various cities of the United States.

Power Station Economics: D. D. Higgins, National Engr., Dec., 1920, p. 563; Jan., 1921, p. 1.

Uniform Costs for Power Plant: Alfred Baruch, Power Plant Engrg., June 15, 1923, p. 623.

Financial Engineering: O. B. Goldman, John Wiley & Sons.

356. Permanent Statistics. — Tables 108 to 110 are taken from the records of a large isolated station in Chicago and serve to illustrate the makeup of the "permanent statistics." The complete file covers each item of equipment and includes the various drawings, specifications, and guarantees for the entire mechanical equipment. Since these records do not vary with the operation of the plant, they require no further attention, once they are compiled, except of course for such changes as may be made from time to time in the plant itself.

357. Operating Records. — The operating records of any plant bear the same relationship to the economical operation of that plant as the bookkeeping and cost accounting systems bear to the manufacturing plant. The distribution of profit and loss in either case can only be obtained by itemizing the various factors involved and by grouping them in such a manner as to show at any time where improvement is possible. Commercial bookkeeping has been more or less standardized and entails very little need of originality on the part of the bookkeeper, but the selection and maintenance of a system of power-plant records may require considerable study and experimenting, since each installation is a problem in itself. The items included in the different forms depend upon the apparatus provided for weighing and measuring the coal and water, the type and number of instruments available for measuring temperature, pressure, and power, and the system adopted for keeping track of oil, waste, general supplies, and repairs. In large stations, autographic recording and integrating appliances, which are to be found in nearly all strictly modern stations and represent but a small part of the first cost of the plant, greatly reduce the labor of keeping continuous records. In many small plants, the cost of autographic instruments may prove to be prohibitive and recourse must be had to the usual indicating devices.

TABLE 109

PERMANENT STATISTICS

BOILERS

Make of boiler	Stirling
Total number in plant	5
Date of installation	
Steam pressure, gage	150
Safety-valve pressure	160
Type of safety valve	Pop
Area of grate, sq. ft.	
Heating surface, sq. ft.	3,500
Superheating surface, sq. ft.	None
Number of steam drums	3
Diameter of steam drums, in.	36
Distance between steam drums, ft.	3
Thickness of shell, in	$\frac{3}{4}$
Thickness of head, in.	$\frac{1}{4}$
Diameter of steam nozzle, in.	10
Diameter of safety valve, in.	4
Diameter of blow-off, in.	2 5
Diameter of feed pipe, in.	2
Temperature of flue, deg fahr.	450–490
Temperature of feed water, deg fahr.	210
Ratio of heating surface to grate area	11 6
Kind of fuel	Illinois screenings
Type of grate	Green chain grate
Rated horsepower	375
Number in battery	1
Weight of boiler	62,186
Cost of boiler and fittings (each)	$5,400
Height of setting	17 ft. 9 in.
Length of setting	17 ft. 4 in.
Width of setting	15 ft. 3 in.
Weight of setting, lb.	272,000
Thickness of wall	Side 20 in.; back, 15 in.
No. of bricks, fire	6,590
No. of bricks, common	19,600
Dimensions of foundation	15 ft. 2 in. × 17 ft. 4 in.
Material of foundation	Stone and concrete
Cost of foundation and setting (each)	$1,500
Distance between batteries	4 ft. 6 in.
Distance back of boiler	17 ft. 6 in.
Distance in front of boiler	16 ft. 6 in.
Distance overhead	2 ft. 10 in.
Number of tubes	337
Diameter of tubes, in.	3 25
Length of tubes, ft.	12 to 14
Steam space, cu. ft.	96
Water space, cu. ft.	643
Kind of draft	Natural
Inches of draft in breeching (maximum)	2.5

TABLE 110

PERMANENT STATISTICS

FEED PUMPS

Date of installation	
Make	Snow
Number in plant	2
Height, ft.	3
Length, ft.	12
Width, ft.	4
Weight of pump	5 tons
Cost, each	$965
Steam pressure	150
Back pressure	$\frac{1}{2}$
Number of valves	32
Character of valves.	Rubber, brass lined
Area thro' valve seats, sq. in., per pump	12 13
Gallons of water per min., per pump	800
Pounds of water per 24 hr., average, actual	479,400
Gallons of water per 24 hr.	599
Volume of air chamber, cu. ft.	3
Shop number	24,572–3
Diameter of steam cylinder.	16
Diameter of water cylinder.	10
Stroke	12
Displacement per stroke, cu. ft	0 545
No. of strokes per min., average	12
Diameter of suction	8
Diameter of discharge	5
Diameter of steam pipe	2.5
Diameter of exhaust	4
Diameter of steam drips	$\frac{1}{2}$
Diameter of water drains.	$\frac{1}{2}$
Suction head, lb. per sq. in.	$1\frac{1}{2}$
Discharge head, lb. per sq. in.	175
Kind of piston packing	Outside packed plunger
Size of piston packing	
Kind of rod packing	Soft
Size of rod packing	$\frac{5}{8}$
Temperature of feedwater	208

In the latter case, continuous records may be closely simulated by plotting the readings of the indicating appliances, say every fifteen minutes, or even once every hour, and by connecting the points with a straight line. The shorter the interval between readings, the smaller will be the error, but unfortunately the duties of the operating engineer in the small plant are usually such as to make frequent readings impossible. Total quantities may be obtained by summing up the various items or by integrating the graphical chart by means of a planimeter. It is not sufficient to record monthly or yearly averages. Daily and even hourly records are absolutely essential for maximum economy. The various losses may be reduced to a minimum only by an intelligent analysis of daily records. A number of forms taken from the files of various power plants are reproduced in this chapter under the proper subheadings and serve to illustrate good practice.

Operation of Central Power Stations. J. D. Morgan, Power, Oct. 7, 1919, p. 550.

Record Keeping for Isolated Power Plants National Engr., Mar. 1, 1918, p. 94.

Operating Charts at West Reading Plant of Metropolitan Edison Power Station: Power Plant Engrg., Aug 1, 1923, p. 757

Classification of Accounts and Standard Form of Statement: National Assoc. of Building Owners and Managers, Edison Building, Chicago.

358. Output and Load Factor. — There are so many ways of expressing the " output " and so many kinds of " factors " in the modern operating code that much confusion arises from the different interpretations of these terms. The various national engineering societies have published codes on definitions and values but there is no generally accepted standard. Until such a standard has been established, it is well to define all terms, the meaning of which may be subject to misconstruction, when reporting the performance of a machine, plant, or system.

In the accompanying tables and charts, the term " output " without qualification refers to the net energy generated by the machine, plant, or system, that is, the energy actually available at the source of distribution. For an electric power station this is the gross kw-hr. generated less the kw-hr. used by the station itself.

According to the Standardization Rules of the American Institute of Electrical Engineers, the **load factor** of a machine, plant, or system is the ratio of the *average* power to the *maximum* power during a given period of time. The average power is taken over a certain period of time, such as a day, a month, or a year, and the maximum power is taken as the average over a short interval of the maximum load within that period. In each case the interval or maximum load, and the period over which the average is taken should be definitely specified, such as a " half-hour monthly " load factor. The proper interval and period are usually

dependent upon local conditions and upon the purpose for which the load factor is used.

The following tentative definitions have been published by the N.E.L.A. Prime Movers Committee, 1922.

Station Load Factor. — The ratio of gross station output in kw-hr. during a given period to the product of the maximum load occurring

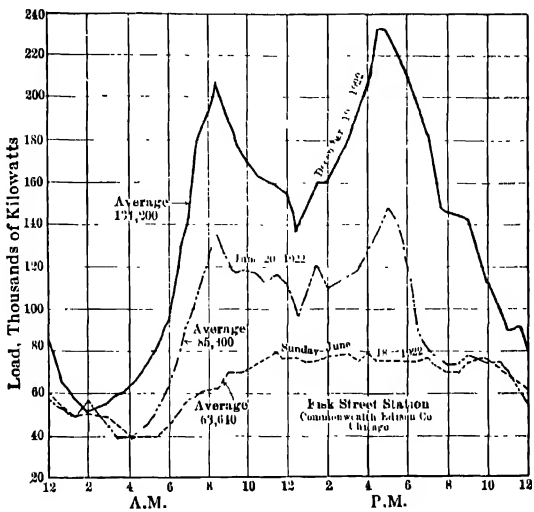


FIG. 671. Typical Daily Load Curve. Large Central Station.

during the period, times the number of hours during that period. In each case, the duration of maximum load and the period over which the station output is measured should be definitely stated.

System Load Factor. — Same as above, except substitute the term "system" for "station."

Station Output Factor. — The ratio of the gross station output during a given period to the sum of the products of the rated capacity (rated capacity is the maximum kw. load which the generating unit can deliver continuously) of each generating unit in the station, by the number of hours it was in operation during that period.

A high output factor, whether based on gross or net output, and whether applied to individual machines, stations, or systems, is essential to maximum economy.

In this text the term "load factor" without qualification is the ratio of the total gross output to the total rated output for a period of one year or 8760 hr.

The *demand factor* is the ratio of the maximum demand to the connected

load. *There is a general tendency to overestimate the maximum electric demand, due, in a measure, to the possibility of all the lights and* motors being in use at one time. Practically speaking, such conditions are not likely to occur. Table 113 gives an idea of the value of the demand factor for various classes of service and may be used as a guide for determining the size of prime movers.

TABLE 111

YEARLY LOAD FACTORS (1922)

Central Stations

Plant	Yearly Output kw.-hr. (Millions)	Peak Load Kw. (Thousands)	Load Factor Per Cent
Blackstone Valley	110	38	39.4
Buffalo Gen. Elec.	587	142	47 3
Cleveland Elec., Ill.	688	181	43 5
Cleveland	116	25	52 3
Consolidated Gas, Baltimore	568	139	46 5
Consumers Power	462	116	45 5
Dayton Power & Lt.	160	46	39 7
Denver Gas & El.	103	29	40 6
Duquesne Light Co.	858	209	46.8
Edison Companies			
Boston	439	133	37 7
Brooklyn	517	164	36 0
Commonwealth	2225	600	47 5
Detroit	1105	255	49 5
Metropolitan	127	32	47 6
New York	1659	497	38 0
Southern Calif.	1199	239	57.2
Toledo	216	58	42 5
Hartford Elec. Lt	134	56	27.3
Indianapolis Lt.	138	38	38 5
Kansas City Power	253	59	49 0
Minneapolis, G E	426	102	47 5
Narragansett El	297	85	39 9
Nebraska Power	140	30	53 4
Niagara Falls Power Co.	2252	329	78 0
North Am. (Mis.)	554	134	47 0
Pacific Gas & Elec.	1609	294	62 5
Penn. Power Co.	495	107	52 8
Philadelphia Elec.	957	260	42 9
Pub. Service Cos.			
Central Ill.	128	31	47 0
New Jersey	958	250	42 1
Northern Ill.	362	83	49 9
Ohio	317	80	45 2
Rochester Gas & El.	193	53	39.4
West Penn.	635	137	53.0
Union Gas & Elec.	366	100	41 8
United Light	129	35	42.1

The *diversity factor* may be defined as the ratio of the sum of the individual maximum demands of a number of loads during a specified period

to the simultaneous maximum demand of all these same loads during the same period. If all the loads in a group impose their maximum demands at the same time, then, the diversity factor of that group will be unity.

Expressed algebraically

$$\text{Diversity factor} = \frac{\text{sum of individual maximum demands}}{\text{maximum demand of entire group}}. \qquad (287)$$

The diversity factor has a very significant bearing on reducing the cost of power; namely, diversity of demand.

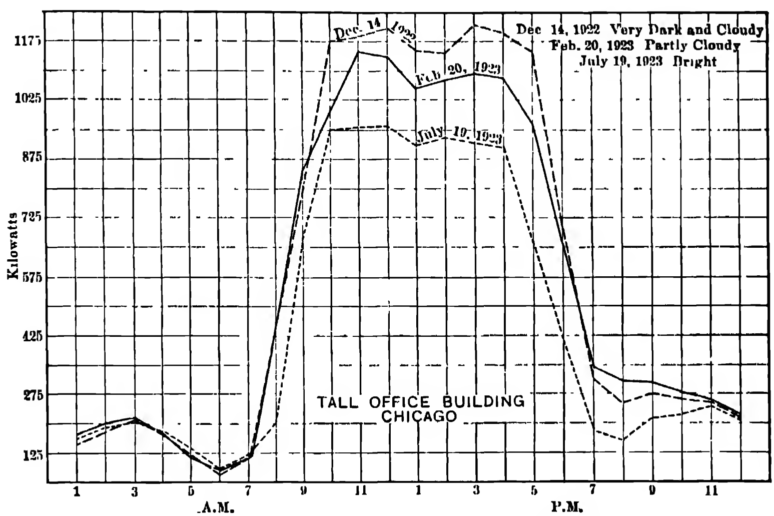


FIG. 672. Typical Daily Load Curve. Tall Office Building, Chicago.

Few central stations are favored with a constant load. The peak of the industrial load occurs during the daytime, that of the railway load for a period in the morning and again in the evening, that of the lighting load for the early evening, etc. Since the maximum demands of the various customers of the different classes do not occur at the same time, the machinery used to supply one class of service at one period may be used to supply other kinds of loads at another, whereas, if the peaks occurred simultaneously the plant would require capacity enough to supply the sum of all the maximum demands of the different customers. The combined maximum demand imposed on the average central station is usually less than half the sum of all the maximum demands of the various customers; therefore, the investment involved in providing the service is approximately half of what it would be if the demands occurred simultaneously.

TABLE 112

DIVERSITY FACTORS AMONG THE DIFFERENT ELEMENTS OF A CENTRAL-STATION DISTRIBUTING SYSTEM

(H. B Gear)

Elements of Distributing System	Diversity Factors			
	Residence Lighting	Commercial Lighting	General Power	Large Users
Among consumers	3.36	1 46	1.44	....
Among transformers	1.30	1.30	1 35	1.15
Among feeders	1 15	1.15	1.15	1 15
Among sub-stations	1 10	1.10	1.10	1.10
Consumer to transformer	3 36	1.46	1.44	....
Consumer to feeder	4 35	1.91	1 95	1.15
Consumer to sub-station	5.00	2 19	2.24	1 32
Consumer to generator	5.53	2 41	2 45	1.45

TABLE 113

CENTRAL STATIONS, DEMAND FACTORS

Demand factors compiled by Commonwealth Edison Company of Chicago

CLASS OF SERVICE

	Demand Factor
Lighting customers:	
Billboards, monuments, and department stores	85 6
Offices	72 4
Residences and barns	60 0
Retail stores	66.3
Wholesale stores	70 1
Average	69.8
Motor customers:	
Offices	65.1
Public gathering places and hotels	28 7
Residences and barns	69 3
Retail stores	61 2
Wholesale stores and shops	58 2
Average	59 4

The values given in Table 112 are fairly typical and may be used for estimating purposes where specific data are not available.

Example 91. — Estimate the maximum demand on the various elements of the generating and distributing system if a group of residence customers are all connected to one transformer, assuming that the maximum peak demand of the group is 42 kw.

Solution. — Using the values in Table 112 we have
Max. demand on

Transformer	= 42	3.36 =	12.5 kw.
Feeder	= 12.5	1.30 =	9.61 kw.
Sub-station	= 9.61	1.15 =	8.36 kw.
Generator	= 8.36	1.1 =	7.6 kw.

Hence, allowing 25 per cent loss between customer's meter and generator, $7.6 \div 0.75 = 10.0 +$ kw. generator capacity must be available at the station to furnish the customers with their maximum demand of current.

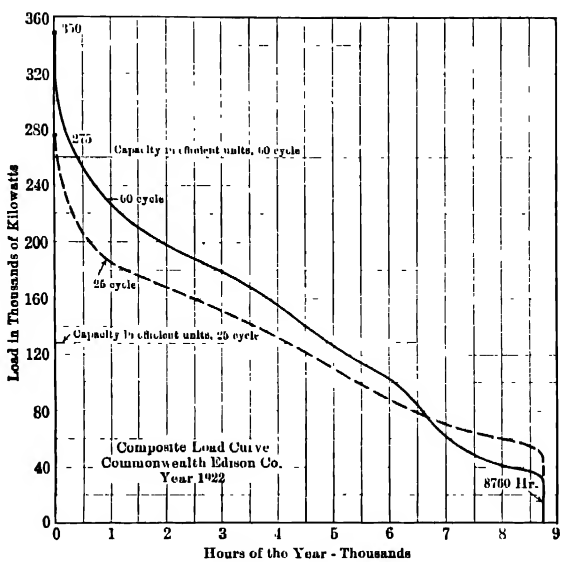


FIG. 673. Composite Load Curve. Commonwealth Edison Co.

The term *power factor* applies to alternating-current generation only and is defined as the ratio of the true power or kw. measured by a wattmeter to the apparent power or kilovolt-amperes (kva.). The actual output in kw., therefore, is equal to the kilovolt-amperes multiplied by the power factor (F). Thus, the i.hp. of an engine required for an alternating-current engine-generator set may be expressed

$$\text{i.hp.} = 1.34 \text{ kva.} \times F \div E \tag{288}$$

in which

E = mechanical efficiency of the entire unit.

Other notations as interpreted above.

The power factor varies with the nature of the electrical load, and varies from 0.95 for a plant in which the load is largely due to the use of synchronous motors or rotary converters to 0.70 where a large part of the station load is due to the use of induction motors, electric furnaces, or arc lighting. In the average large central station generating current for lighting and power, the power factor is approximately 0.80.

Operating Code Definitions: N.E.L.A. Report of Prime Movers Committee, T3, 1922, p. 337.

A.S.M.E Code on Definitions and Values: Mech. Engrg., Sept., 1923, p. 548.

Load Factor; Its Definition and Use: The Canadian Engr., Jan. 20, 1921, p. 149.

Effect of Load Factor on Steam-station Costs: Power, Jan. 4, 1921, p 24.

Diversity and Diversity Factors: Terrell Croft, Power, Feb. 6, 1917, p. 171.

Power Factor Problems in Industrial Plants: Power Plant Engrg., Nov. 1, 1923, p. 1087.

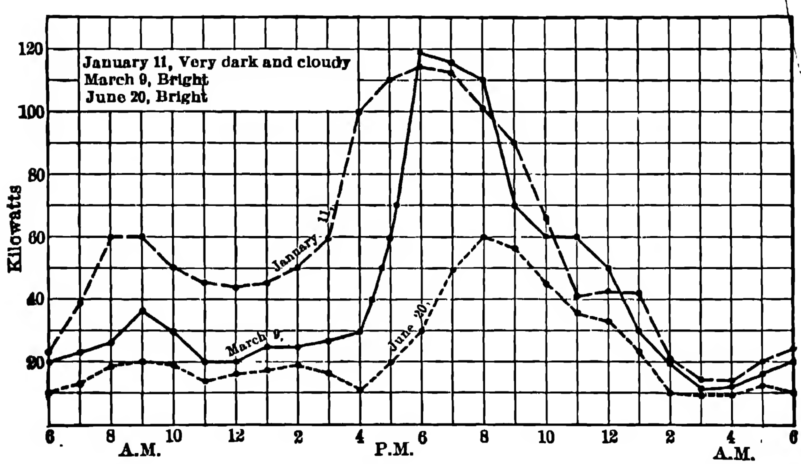


FIG. 674. Typical Daily Load Curve. Large Apartment Building.

In any system the *total* fixed charges per year are constant irrespective of the load factor, since interest, taxes, depreciation, insurance, and maintenance go on whether the plant is in operation or not. The total fixed charges for a specific case are illustrated in Fig. 675 by a straight line. The fixed charges per kw-hr., however, decrease as the load factor increases. Considering the values in Fig. 675, with the plant operating continuously for 8760 hr. at rated load (100 per cent load factor) the fixed charges per kw-hr. are

$$65{,}000/(5000 \times 8760) = \$0.00148.$$

With 30 per cent load factor these charges are

$$65{,}000/0.3(5000 \times 8760) = \$0.00495 \text{ kw-hr.}$$

The higher the load factor, the greater is the amount of power produced and the longer does the apparatus work at best efficiency. But the greater the power produced, the larger will be the fuel consumption and the oil and supply requirements. The labor charges will be practically constant. The total operating cost per year increases as the load factor

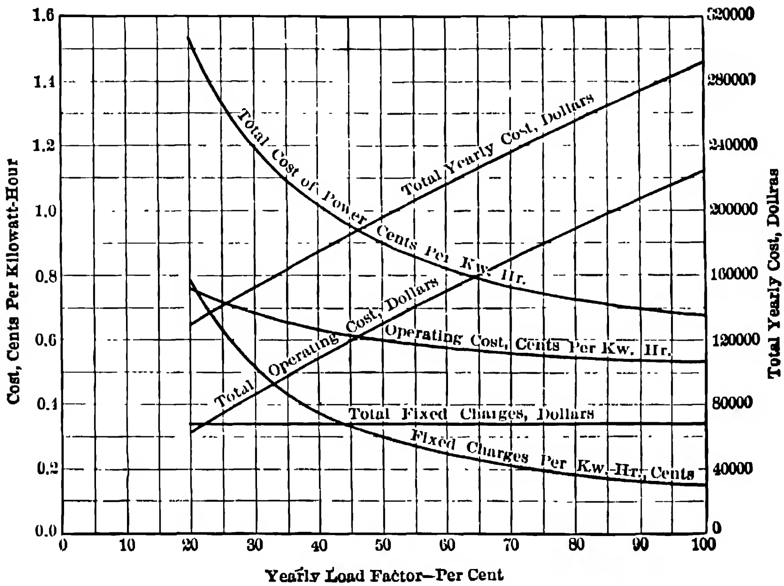


FIG. 675. Influence of Load Factor on Cost of Power at Switchboard (Maximum Load 5000 Kw.).

increases, but not directly. (See Fig. 675.) The cost per kw-hr., however, decreases as the load factor increases. For example, the operating costs per year with plant operating continuously at full load are \$230,200. This gives

$$230{,}200/(5000 \times 8760) = \$0.00525 \text{ per kw-hr.}$$

With 30 per cent load factor the total yearly operating charges are \$87,980, which gives

$$87{,}980/0.3(5000 \times 8760) = \$0.0067 \text{ per kw-hr.}$$

In general, the higher the load factor, the greater becomes the ratio of the operating to the fixed charges, and extra investment may become advisable to secure the greatest possible economy.

On the other hand, when the load factor is low the fixed charges are the governing factor in the cost of power, and extra expenditures must be carefully considered, particularly if fuel is cheap.

The total fixed charges are frequently as great if not greater than the actual operating costs, and must necessarily be included in arriving at the total overall cost.

359. Cost of Power. General. — The actual cost of producing power depends upon the geographical location of the plant, cost of fuel and labor, the size of apparatus, the design, conditions of loading system, of distribution and the method of accounting. Comparisons based on the cost per hp-hr. or per hp-yr., or the equivalent are without purpose because of the many variables entering into the problem. It is impossible to intelligently compare costs or to obtain a true understanding of what costs for power really mean without a thorough knowledge of the various items entering into the unit cost such as costs of fuel, oil, waste, repairs, labor, insurance, taxes, management, distribution, maintenance and allowance for depreciation. In addition to these an understanding must be had of the operating conditions, such as size of plant, load factor, variation in load, ratio of the maximum load to the economic full load, number of hours a day the plant is operated and the like. With each plant having an individuality distinctly its own, in so far as the charges which go to make up the ultimate cost is concerned, it is practically impossible to arrive at any definite conclusion as to the manner in which the real cost of power may be correctly determined for purposes of comparison. Perhaps the best method of stating station economy is to give the average yearly heat units supplied by the fuel per kw-hr. delivered to the switchboard, and the load factor. This eliminates price and quality of fuel and offers a fairly satisfactory criterion of the efficiency of operation.

In any case the cost of power is based upon the expense which is independent of the output, or *fixed charges*, and that which is a function of the output, or *operating costs*. In the small plant the items included in the fixed and operating costs are comparatively few in number and require but an elementary knowledge of bookkeeping, but in large industrial organizations or central stations the number of separate items to be considered may run into the hundreds and necessitate a complex system of accounting. Some idea of the different systems employed with examples of cost of power in specific cases may be gained from an inspection of Tables 124 to 134.

360. Fixed Charges. — These cover all expenses which do not expand and contract with the output. In the privately owned plant the fixed charges are usually limited to interest on the investment, rental, depreciation, taxes, insurance and sometimes maintenance, though the latter is ordinarily included in the operating costs. The accounting systems for public electric light and power companies are usually prescribed by the Public Utility Commission of the state in which the plant is located and

the various charges must necessarily conform with the rules formulated by this Commission.

361. Interest. — The rates of interest on borrowed money vary with the nature of the security. In the case of small power plants, the form of security is usually a first mortgage bond on the plant and equipment. In the larger plants, the money or credit may be obtained through the sale of stocks, collateral notes, debenture bonds, and other classes of securities. If a builder has sufficient funds to construct the plant without borrowing, he should charge against the item "interest" the income which the sum involved would bring if placed out at interest or if conservatively invested in his own business. In estimating the interest

TABLE 114

COST OF MECHANICAL EQUIPMENT — STEAM TURBO-ELECTRIC GENERATING STATIONS
60,000 Kw Capacity
(1923)

	Dollars per Kw	
	High	Low
Preparing site — Dismantling and removing structures from site, making construction roads, tracks, etc.	$0 50	$.
Yard Work — Intake and discharge flumes for condensing water, railway siding, grading, fencing sidewalks	8 00	6 00
Foundations — Including foundations for building, stacks, and machinery, together with excavation, piling, waterproofing, etc.	10 00	8 00
Building — Including frame, walls, floors, roofs, windows and doors, coal bunker, etc., but exclusive of foundations, heating, plumbing, and lighting	15 00	12.00
Boiler-room Equipment — Including boilers, stokers, flues, stacks, feed pumps, feedwater heater, economizers, mechanical draft, and all piping and pipe covering for entire station except condenser water piping	35 00	28 00
Turbine-room Equipment — Including steam turbines and generators, condensers with condenser auxiliaries and water piping, oiling system, etc.	30 00	25.00
Electrical Switching Equipment — Including exciters of all kinds, masonry switch structure with all switchboards, switches, instruments, etc., and all wiring except for building lighting . . .	15 00	12.00
Service Equipment — Such as cranes, lighting, heating, plumbing, fire protection, compressed air, furniture, permanent tools, coal- and ash-handling machinery, etc.	14 00	10 00
Starting Up — Labor, fuel, and supplies for getting plant ready to carry useful load	1 50	1 00
General Charges — Such as engineering, purchasing, supervision, clerical work, construction plant and supplies, watchmen, cleaning up	8 00	6 00
Total cost of plant to owner, except land and interest during construction	$137 00	$108 00

charges, 6 per cent of the capital invested is ordinarily assumed unless specific figures are available. Initial costs for various types of plants are to be found in the accompanying tables, but so much depends upon the grade of equipment, market price of materials, cost of labor, and plant location that these figures are only of academic value.

Cost of Money to Utilities: Elec. Wld., Sept. 15, 1923, p. 538.
Power Plant Accounts — Interest: Power, Oct. 25, 1921, p. 636.

TABLE 115

APPROXIMATE AVERAGE COST OF MODERN STEAM TURBO-ELECTRIC PLANTS (CONDENSING)

(1923)

Units and Auxiliaries Installed and Erected	Size of Plant — Kw.				
	500	1000	2500	5000	10,000
Bldg. real estate, excavating.	$60 00	$55 00	$46 00	$40 00	$36.00
Turbo-generators	40 00	35 00	30 00	25 00	23 00
Condensing equipment	25 00	20 00	15 00	10 00	8.50
Boilers, stokers, stacks	55 00	50 00	45 00	42 00	40 00
Bunkers and conveyors	10 00	10 00	8 50	8 00	7 00
Boiler feed and service pumps, heaters	5 00	4 00	3 00	2 00	2.00
Switchboard and wiring	9 00	8 00	7.00	7 00	6 50
Exciters	1 00	3 00	2 50	2 00	1 50
Piping	9 00	10.00	11 00	12 00	13 00
Superintending, engrg., contingencies, etc.	23 00	20 00	17 00	18 00	17 50
Total	$240 00	$215 00	$185 00	$166 00	$155.00

Each plant is provided with a spare boiler and such extra apparatus as is consistent with good practice. Boiler pressure 175 lb. gage. Superheat 125 deg. fahr.

Average figures of this nature are apt to be misleading when applied to any particular case, because of the wide variation in the individual characteristics of each plant. They are intended merely as a rough guide to the variation in cost with size.

362. Maintenance. — Maintenance usually refers to the expense of keeping the plant in running order over and above the cost of attendance, although the term is frequently used in place of "repairs." It includes cost of upkeep, replacement, and precautionary measures. The last item includes the renewal of working parts, painting of perishable or exposed material, and replacing worn-out and defective material. Many engineers make no allowance for maintenance in the fixed charges and include these costs under supplies, attendance, or repairs. In a general way, when maintenance is included under the fixed charges, an annual charge of 2 per cent is considered a liberal allowance, since most of the repair

work comes under attendance. In central station practice, maintenance is divided among the several parts of the system as follows: Buildings, steam appliances, electrical equipment, and miscellaneous. In this connection the maintenance becomes a part of the operating charges, since the various items vary widely from month to month.

363. Taxes and Insurance. — Taxes vary from a fraction of one per cent to 3 per cent, depending upon the location of the plant. An average figure is 2 per cent of the actual value of the investment. Buildings and machinery are ordinarily insured for fire loss, and boilers, compressor tanks, and pressure vessels against accidental explosions. Accident policies are sometimes carried on all operating machinery. A fair charge for this item is one-half per cent.

TABLE 116

COST OF MECHANICAL EQUIPMENT*

F.O.B. FACTORY

(1924)

BOILER ROOM

	Cost per boiler horsepower
Air preheaters	$ 5 00–$10 00
Ash gates, power dumps, hoppers, and gates	4 00 - 5 00
Blowers:	
Multivane, motor-drive, automatic regulation	2 00– 3 00
Undergrate turbo blower	0 75– 1 00
Vano	1 00– 1 50
Boilers:	
Straight tube, 250 lb. pressure and under	20 00– 30 00
Curved tube, 250 lb. pressure and under	18 00– 20 00
Straight tube, 250 lb. pressure and over	30 00– 75 00
Horizontal return tubular	12 00- 18 00
Fire box, locomotive type and vertical tubular	30 00– 40 00
Boiler insulation	0 20– 0.30
Boiler settings:	
Horizontal return tubular	6 00– 9 00
Water tube, low head room	6 00– 12.00
Water tube, high head room	10 00– 15 00
Chimneys:	
Concrete	5.00– 10.00
Radial brick	3.00– 8.00
Self-sustaining steel	3 00– 9.00
Chimney insulation, asbestos	2 50– 4 50
Economizers	14.00– 25.00
Feed pumps:	
Centrifugal	0.25– 1 00
Reciprocating	0.40– 0.60

* Courtesy of Himelblau and Agazim, Chicago.

Feedwater heaters:	
Closed	0.90– 1 50
Open	0.40– 1.50
Feedwater regulators	0.25
Soot blowers	0.50– 1.00
Stokers:	
Hand, with hopper feed	4.25– 9.25
Hand, without hopper feed	3.00– 5.00
Reciprocating	8 00– 10 00
V-type, natural draft, engine, auto., etc.	10 00– 16.00
Traveling grate, natural draft, engine, shafting, etc.	20 00– 25 00
Traveling grate, forced draft, engine, shafting, etc.	40 00– 50 00
Underfeed, single retort complete	9.00– 13.00
Underfeed, multiple retort, engine drive, shafts, etc.	12.50– 18 50
Underfeed multiple retort, complete with rotary power dump	17.00– 23.00
Superheaters:	
Curved-tube boilers	4.00– 7.00
Horizontal-return tubular boilers	4.00– 9 00
Straight-tube boilers	3 50– 12.00

ENGINE ROOM

Turbines, steam	16.00– 30 00 per br. hp.
Turbo-alternators	18.00– 32 00 per kw.
Engines:	
Corliss, simple	25.00– 30 00 per i.hp.
Corliss, compound	35 00– 40 00
Corliss, non-releasing	16 00– 40.00
Poppet valve	16 00– 33 00
Uniflow	20.00– 45.00
Condensers, jet (cost per 1000 lb. steam)	200.00–500 00
Condensers, surface, cost per sq. ft.	2.60– 4 50

364. Depreciation. — Depreciation may be defined as a decrease in value occasioned by wear or age, change of conditions rendering the plant inadequate for its particular functions, or change in the art rendering it obsolete as compared with recent installations. Depreciation may be conveniently classified as:

Natural depreciation, or the gradual decrease in value occasioned by wear and age. This may be largely offset by maintenance.

Functional depreciation due to **obsolescence, inadequacy** or **destruction** by any cause. A thing is obsolete when it has been rendered valueless as the result of change in the art, and this may occur where no physical deterioration has taken place. A thing is inadequate when it is incapable of fully performing the function for which it is intended. Inadequacy indicates neither physical depreciation nor obsolescence; it may result

from expansion of markets, community growth and the like. Obsolescence, inadequacy, and destruction cannot be predicted and charges against this class of depreciation are naturally conjectural.

The term "depreciation" is frequently used when the term "amortization" would be more appropriate. *Amortization* refers to the retirement of invested capital, while depreciation is loss of value.

TABLE 117

APPROXIMATE COST OF RADIAL BRICK STACKS

(1923)

Height, Ft.	Diam., Ft.	Cost	Height, Ft	Diam., Ft.	Cost
75	4′0″	$1700 00	175	8′0″	$ 7,250 00
75	6′0″	2100 00	175	10′0″	9,200 00
75	8′0″	2250 00	175	12′0″	10,500 00
75	10′0″	2700 00	175	14′0″	12,500 00
125	6′0″	3800 00	200	8′0″	9,500 00
125	8′0″	4550 00	200	10′0″	10,300 00
125	10′0″	5300 00	200	12′0″	13,200 00
125	12′0″	6900 00	200	14′0″	15,000 00
150	8′0″	6700 00	250	10′0″	18,100 00
150	10′0″	7100 00	250	12′0″	19,900 00
150	12′0″	8200 00	250	14′0″	21,000 00
150	14′0″	9500 00	250	16′0″	23,000 00

Costs of common brick chimneys, 3/4 lined, are approx. 1.3 times the tabular values.

Reinforced concrete chimneys up to 125 ft. in height by 5 ft. in diameter cost about the same as radial brick; 150 ft. in height by 7 ft. in diameter about 12 per cent less; 200 ft. by 10 ft. about 20 per cent less and 250 ft. by 10 ft. about 25 per cent less.

Costs of full-lined self-supporting steel stacks about 1.1 to 1.2 times the tabular values.

There are several methods of dealing with depreciation; among the more common may be mentioned:

(1) Charging to earnings in good years and crediting to amortization reserve such amounts as the profits from operation permit.

(2) Charging to earnings the amortization as it matures and necessitates renewals.

(3) Charging to earnings and crediting to amortization reserve annually a certain percentage of the cost determined by the average weighted life of the property.

In central-station practice, it is customary to establish a **reserve fund** to allow for depreciation, based on the original cost of the property less

salvage or junk value, spread over a period of years approximating the useful life of the plant. If depreciation is considered to include the maintenance which is charged to expense directly, it would be proper to set aside as a reserve a fixed percentage of the decreasing value of the plant to represent the unmatured decadence. This ideal situation would equalize the total burden over the life by making the depreciation allowance largest when the repairs are smallest, and conversely the depreciation allowance smallest when the repairs are largest at the end of the useful life of the plant. If the system were composed of many small units not requiring renewal at or near the same time, no special reserve would be necessary, as all replacements could be charged directly to operating expenses because these amounts would be inconsiderable in any one year. In the large central station, however, a considerable portion of the plant is composed of large units which the rapid development of the art and growth of business may render obsolete long before their natural life has expired. As a result of and to provide for this condition, depreciation reserves are accumulated either by the "straight-line" or "sinking-fund" method.

TABLE 118

FUNCTIONAL LIFE OF VARIOUS PORTIONS OF STEAM POWER PLANT EQUIPMENT

	Years		*Years*
Belts	8–15	Generators, a.c.	20–35
Boilers, fire tube	20–35	Generators, d.c.	20–35
Boilers, water tube	25–40	Heaters, closed	20–40
Buildings, masonry	25–60	Heaters, open	25–50
Buildings, wooden or sheet iron	15–30	Motors	20–35
Chimneys, masonry	30–60	Piping, exposed	10–20
Chimney, self-sustaining steel	25–50	Piping, protected	15–35
Chimneys, sheet iron, guyed	8–15	Pumps	20–40
Coal conveyors, belt	8–20	Rotary transformers	20–30
Coal conveyors, bucket	10–25	Stokers, chain-grate	20–30
Condensers, jet	25–50	Stokers, underfeed	20–30
Condensers, surface	20–40	Storage batteries	10–20
Economizers, cast-iron	20–30	Transformers	15–30
Economizers, steel	10–20	Turbines, steam	20–40
Engines, high-speed	20–40	Wiring	10–20
Engines, low-speed	25–50	Composite plant	20–30

NOTE. — So much depends upon the design and the conditions of operation that average values for actual physical life are without purpose. Practice shows that most power-plant appliances become obsolete or inadequate long before the limit of their physical life is reached. The values above are purely arbitrary but serve to show the range in functional life assumed by various companies in establishing amortization annuities.

TABLE 119

DEPRECIATION ANNUITY

(Per Cent of First Cost)

Assumed Useful Life of Apparatus	Rate of Interest, Per Cent												
	2	2.5	3	3.5	4	4.5	5	5.5	6	7	8	9	10
2	49 50	49 37	49 27	49 14	49 02	48 90	48.78	48 66	48 54	48 31	48 07	47 84	47.62
3	32 67	32.51	32 35	32 19	32 03	31 88	31 72	31 56	31 41	31 10	30 80	30 51	30.21
4	24 26	24 08	23.90	23 72	23 55	23 38	23 20	23 03	22 86	22.52	22 19	21 86	21 55
5	19.21	19.02	18.83	18 65	18 46	18 28	18 10	17 91	17 71	17 39	17 01	16 71	16 38
6	15 85	15.65	15 46	15 26	15 08	14 89	14.70	14 52	14 34	13 98	13 63	13 29	12 96
7	13.45	13.25	13 05	12 85	12 66	12 17	12 28	12 09	11 91	11 15	11 20	10 87	10 54
8	11.65	11 44	11 24	11 05	10 85	10 66	10 17	10 28	10 10	9 75	9 10	9 06	8 74
9	10 25	10 04	9 84	9 61	9 15	9 26	9 07	8 88	8 70	8 35	8 00	7 68	7 36
10	9 13	8 92	8 72	8 52	8 33	8 11	7 95	7 76	7 59	7 23	6 90	6 58	6.27
11	8 22	8 01	7 81	7 61	7 11	7 22	7 01	6 86	6 68	6 33	6 00	5 69	5.40
12	7 45	7.25	7 04	6 85	6 65	6 46	6 28	6 10	5 93	5 60	5 27	4 97	4.68
13	6 81	6 60	6 40	6 20	6 01	5 83	5 61	5 47	5 29	4 96	4 65	4.36	4.08
14	6 26	6 05	5 85	5.65	5 17	5 28	5 10	4 93	4 76	4 43	4 13	3.84	3.58
15	5 78	5 57	5 38	5 18	4 99	4 81	4 63	4 46	4 29	3 98	3 68	3.40	3.15
16	5 36	5 16	4 96	4 77	4 58	4 40	4 22	4 06	3 89	3 58	3 30	3 03	2 78
17	5 00	4.79	4 59	4 10	4 22	4 01	3 87	3 70	3 54	3 24	2 96	2 71	2 47
18	4 67	4 46	4 27	4 08	3 90	3 72	3 55	3 39	3 23	2 94	2 67	2.42	2 19
19	4.38	4 17	3 98	3 79	3 61	3 41	3 27	3 11	2 96	2 67	2 41	2.17	1 95
20	4 11	3 91	3 72	3 53	3 36	3 19	3 02	2 87	2 72	2 44	2 18	1.95	1.75
25	3 12	2 92	2 74	2 56	2.40	2 24	2 09	1 95	1.82	1 58	1 37	1 18	1.02
30	2 46	2 27	2 10	1 94	1 78	1 64	1 50	1 38	1 26	1 06	0 88	0 73	0.61
35	2 00	1 82	1 65	1 50	1 36	1 23	1 10	1 00	0 90	0 72	0 58	0 46	0.37
40	1 65	1 48	1 33	1.18	1 05	0 93	0 83	0 73	0 64	0 50	0 38	0 29	0.22
45	1.39	1 23	1.08	0 94	0 83	0 72	0 62	0 54	0 47	0 35	0 26	0.19	0 14
50	1.18	1 03	0 89	0 76	0 65	0 56	0 48	0 40	0 34	0 25	0 17	0 12	0 09

Straight-line Method. — This method is based on the assumption that if the total investment, less salvage, is divided by the functional or assumed life of the plant, the resulting quotient expresses the amortization installment or the amount which should be allowed each year to cover the accrued amortization. This is the simplest of the several methods that have been suggested for calculating depreciation annuities with which to establish depreciation funds, and for short-lived plants it offers a fairly satisfactory means of estimating the depreciated value. The straight-line method is shown graphically in Fig. 676. The original cost is composed of the net cost of labor and material plus the overhead (the extra charge intended to cover engineering and architectural fees, fire and liability insurance, and interest on the investment during construction; contractors' profits on the portion of the work not done by the company itself; legal organization and incidental expense). The **functional life** of the plant is a purely arbitrary quantity and is supposed to represent the weighted average period of usefulness of the various units com-

posing the plant. The **actual physical life** of the various units composing the plant can only be approximated, since everything depends on the grade of material, workmanship, and upkeep. Most power-plant appli-

TABLE 120

TYPICAL OPERATING CHART

DAILY POWER-HOUSE REPORT

THE UNITED LIGHT AND POWER CO.

.. .. Division

....19..

Weather — Noon

							Hr.	Min.
Engine No. 1	started	... M	stopped	... M	Total time run	..	....
Engine No. 2	started	. M	stopped	M	Total time run	..	......
Inc current on		M	off	....M	Total time on	....	.....
Street arcs on		M	off	. . M	Total time on	.	.....

Noon AMPERE READINGS

12 00	12 30	1 00	1 30	2 00	2 30	3 00	3 30	4 00	4 15	4 30	4 45	5 00	5 15	5 30	5 45	6 00	6 15	6 30
6 45	7 00	7 15	7 30	7 45	8 00	8 15	8 30	8 45	9 00	9 15	9 30	9 45	10 00	10 30	11 00	11 30	12 00	1 00
2 00	3 00	4 00	5 00	5 15	5 30	5 45	6 00	6 15	6 30	6 45	7 00	7 15	7 30	7 45	8 00	9 00	10 00	11 00

Coal used . ... . lb	**Coal Received on Track.**	**Boilers in Service.**
Cylinder oil . . . pt.	Car No.	No 1 from m to . .. ..m
Engine oil ... . pt.	Initial	No. 2 from m to . .....m
Waste . ... lb	Time placed . ... m	No 3 from m to . .... m
Water . . . .. cu. ft.	Time released ... . m	Washed No. . . ..
Carbons . .	Weight . . .. .lb.	Blew No. .. ..
Globes outer inner .	Ashes sold . .. .loads to	. . ..

Material Received for Power House Use..	Total Kilowatt Output Read meter 12 o'clock noon
....	
....	Meter to-day .. . .Kw.
....	Meter yesterday . . ...Kw.
. . ..	Diff. . . ..

Report here ANY interruption of service either arc or incandescent.

Time off .. ..Cause . . . .

Arc lights out .. ..

Lights .. . .

Location Reported by

ances become obsolete or inadequate long before the limit of their physical life is reached. If the actual physical life could be accurately determined and the annual cost of repairs and upkeep were uniform, the straight-line method would be accurate for calculating the depreciated value; but since this is seldom, if ever, the case, this method should be used only

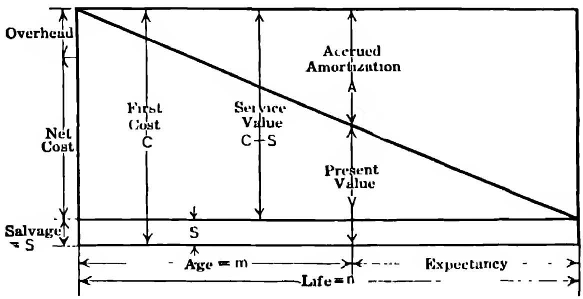


FIG. 676. Showing Straight-line Method of Amortization.

for determining depreciation annuities. The various values on the diagram in Fig. 676 may be expressed algebraically, thus

$$D = (C - S)/n \tag{289}$$

$$V = S + (C - S)\frac{n - m}{n} \tag{290}$$

$$A = C - V \tag{291}$$

in which

D = amortization installment or depreciation annuity,
C = first or original cost,
S = salvage value,
A = accrued depreciation annuity,
V = the depreciated or present value,
m = age of the plant,
n = functional life.

Sinking-fund Method. — According to this method of calculating depreciation annuities, it is assumed that the accrued depreciation of the property is the amount already accumulated in a **sinking fund** that was begun when the property was first put into service, and the annuities of which are such that at compound interest the amount at the end of the functional life of the property will equal the first cost. The various factors entering into the problem as shown in Fig. 677 are based on the following equations, which may be found in text books on compound interest.

The value of 1 placed at compound interest (rate $= r$) at the end of n years is

$$(1 + r)^n. \tag{292}$$

The amount of an annuity of 1 in n years is

$$[(1 + r)^n - 1] \div r. \tag{292a}$$

The present value of an annuity of 1 for n years is

$$[(1 + r)^n - 1] \div r(1 + r)^n. \tag{292b}$$

The annuity which 1 will purchase for n years is

$$[r(1 + r)^n] \div [(1 + r)^n - 1]. \tag{292c}$$

The annuity which would amount to 1 in n years is

$$r \div [(1 + r)^n - 1]. \tag{292d}$$

Applying these fundamental equations to the values in Fig. 677, we have

$$D = (C - S)r \div [(1 + r)^n - 1]. \tag{293}$$

$$V = S + (C - S)\left[1 - \frac{(1 + r)^m - 1}{(1 + r)^n - 1}\right] \tag{294}$$

$$A = C - V. \tag{295}$$

All notations as in equations (289) to (291).

The sinking-fund method is applicable to calculation of the depreciated value of a property only where *natural* depreciation is involved and where current repairs are uniform or absent.

In establishing a sinking fund, it is not supposed that an owner will regularly lay aside an annual amount, or take the trouble to arrange for its investment at current rates in the market or savings bank, since the money is probably worth more to him in his business. In practice, it is retained in his business or investments and is earning the rate of interest obtainable therein; but in determining the net profit or loss this depreciation item is nevertheless accounted for just as if it were actually placed in outside investments.

The **expectancy** or remaining life of any article is the probable time during which it may reasonably be expected to render efficient service. It is determined from the actual condition of the article and all local circumstances which may affect its continued use, and not by subtracting age from probable life. Thus an article may have a probable life of twenty-five years and yet be in first-class condition and as good as new when it reaches the end of this term. The value of this article is not written off the books nor should it be considered as good as new. Its value

is ascertained by determining its probable additional years of usefulness and the probable cost of replacing it at the end of this term.

TABLE 121

TYPICAL OPERATING CHART

(Large Chicago Department Store)

. . ..19..

Monthly Report

Date	Average Outside Temperature	Fuel						Supplies			
		Coal				Ash		Oil Used, Gal.		Waste, Lb.	Total Water to Building, Cu. Ft.
		Kind	Lb Burned	Cost Per Ton	Cost Per Day	Lb. Removed		Engine	Cylinder		

Output				Engine-Hr Run					Boilers-Hr. Run						Breeching	
Boilers		Generators														
Lb of Water Evaporated	Water Evaporated Per Lb. of Coal	Amp. Hr.	Kw.-Hr.	1	2	3		5	1	2	3	4	5	6	Draft	Temperature

Heating System		Ventilating Plants, Hr. Run		Refrigerating Plant			Repairs-Hr.		
Steam Pressure	Live Steam-Hr.	Fan 1	Fan 2	Hr. Run	Gas Used, Lb.	Ice Made, Lb.	Engine Room	Boiler Room	Miscellaneous

In the original copy all of these items are conveniently grouped on one large form ruled for 31-day entries with space at bottom for total quantities and costs. In the reproduction only the headings are included.

The **replacement** may mean the substitution of a new plant exactly similar to the old one, but at a cost dependent on new conditions, new prices of labor and material, or it may mean the substitution of new devices rendering equivalent service. In either event the replacement may be at a greater or less cost than the original cost, with, therefore, a corresponding increase or decrease of capital invested. Expenditures for

TABLE 122

TYPICAL OPERATING CHART

THE EDISON ILLUMINATING CO., *Detroit, Mich.*

TURBO-GENERATOR LOG

DELRAY POWER HOUSE No. 2
For 24 hours ending midnight.
Day Date

	Turbine No 6	Turbine No. 7	Turbine No 8	Turbine No 9
Started	*12-55*, *6-19*	*6-12*	*7-18*	*5-14*
Stopped	*11-32*		*9-20*	*12-15*
Running	*17* hr. *46* min.	*17* hr. *48* min.	*14* hr *2* min.	*18* hr. *59* min.
Idle	" *10* "	" *22* "	" *15* "	" *22* "
Repair	" "	" "	" "	" "

Engineer on Watch

6:30 A.M. to 2:30 P.M. *Wm. Crawford*

2 30 P.M. to 10:30 P.M. *J. Marcth*

10 30 P.M. to 6.30 A.M. *T. Callaghan*

Hour	Barometer	Turbine No 6 Pressures Throttle	1st Stage	Step	Condenser in Abs.	Turbine No. 7 Pressures Throttle	1st Stage	Step	Condenser in Abs.	Turbine No 8 Pressures Throttle	1st Stage	Step	Condenser in Abs.	Turbine No 9 Pressures Throttle	1st Stage	Step	Condenser in Abs.	Kilowatts Load Power House No. 1	Power House No. 2	Total	Remarks
A.M. 1	29 50	197	31	925	0 9													10 000	8 000	1[illegible] 000	
2		200	42	950	0 9													7 000	8 900	1[illegible] 900	*No. 9 turbine examined — there is no trace of oil working through anywhere.*
3		197	39	945	0 9													8 000	8 800	1[illegible] 800	
4		196	40	948	0 9													6 800	8 800	17 600	
5		193	42	950	0 9													9 800	9 200	1[illegible] 000	
6		200	42	950	1 0									194	40	975	0 9	12 800	19 400	3[illegible] 200	
7	29 50	197	47	955	1 0	197	47	870	0 9					200	50	980	1 0	23 900	30 800	54 700	*No. 9 Circulating engine and circulator. All bearings.*
8		192	45	950	1 0	194	47	870	0 9	194	53	970	10	196	43	975	0 9	21 900	41 000	62 900	

TABLE 123

TYPICAL OPERATING CHART

THE EDISON ILLUMINATING CO., *Detroit, Mich.*

DELRAY POWER HOUSE No. 2

For 24 hours ending midnight

CONDENSER ROOM LOG

Day Date

Hour	Unit No. 6				Unit No. 7				Unit No. 8				Unit No. 9				No 8 Elect B.F.P.	Remarks
	R.p.m. D.V. Pump	*63*	R.p.m Circul. Pump	*160*	R.p.m D.V. Pump	*63*	R.p.m Circul Pump	*162*	R.p.m. D.V. Pump	*66*	R.p.m. Circul Pump	*151*	R.p.m D.V Pump	*63*	R.p.m Circul. Pump	*162*		
	Water Temperatures				Water Temperatures				Water Temperatures				Water Temperatures					
	Entering Condenser	Leaving Condenser	Leaving Wet Pump	Leaving B.F. Pump	Entering Condenser	Leaving Condenser	Leaving Wet Pump	Leaving B.F. Pump	Entering Condenser	Leaving Condenser	Leaving Wet Pump	Leaving B.F. Pump	Entering Condenser	Leaving Condenser	Leaving Wet Pump	Leaving B.F. Pump		
A.M.																		
1	*39*	*49*															*156*	*No 7 condenser*
2	*39*	*50*															*162*	*washed, tested.*
3	*39*	*50*	*63*														*166*	*Ferrel repacked.*
4	*39*	*50*	*62*														*149*	
5	*38*	*51*	*64*														*131*	
6	*38*	*51*											*38*	*50*			*170*	
7	*38*	*52*	*64*	*192*	*38*	*52*	*62*					*168*	*38*	*52*	*62*			
8	*40*	*53*	*64*	*190*	*40*	*54*	*64*		*40*	*53*	*60*	*172*	*40*	*53*	*62*			

new parts of a plant, which take the place of old parts retired for any cause, should be charged to replacement only to the extent of capital represented by the part of the plant thus retired. Any excess of the expenditure for replacement over the cost of the discarded part of a plant should be treated as an addition to, and any less cost as a deduction from, the invested capital. The term " replacement " should not be used in the sense of retirement of invested capital, which refers to the cost of the replaced part and not to the cost of the new equivalent installation. (" Valuation, Depreciation, and the Rate-Base," Grunsky, John Wiley & Sons, 1917.)

The term **going value** may be properly taken to mean a value attached to a public utility property as the result of its having an established revenue-producing business. Going value may be determined from a

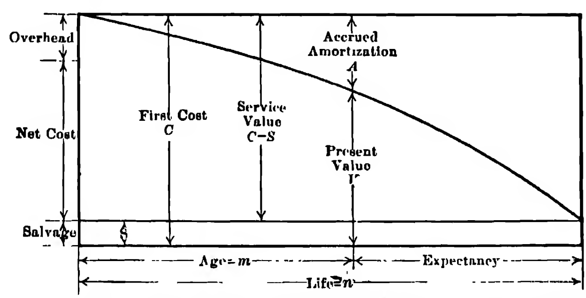


FIG. 677. Showing Sinking-fund Method of Amortization.

consideration of the amounts of money actually expended in the cost of producing the business or it may be determined from consideration of the present cost of reproducing the present revenue. (" Value for Rate-Making," Floy, 1917.)

For purpose of design and comparison, it is customary to assume a single fixed percentage for depreciation, obsolescence, inadequacy, etc. An average figure is 5 per cent.

Unit-cost Depreciation. — The unit-cost depreciation formulas, as developed by H. P. Gillette,[1] are rational and applicable to all problems involving depreciation, although the data for accurate application may not always be available.

Let C and c = first cost of the new and the old property, respectively,
F and f = functional depreciation annuity rates for the new and old plants, respectively,
r = interest rate including risks, taxes and management, not included in F, f, E and e.

[1] " Mechanical and Electrical Cost Data," 1918, p. 99–103.

R = interest rate plus functional depreciation = $r + f$ or $r + F$,
S and s = salvage value of the new and of the old property, respectively,
E = equated annual operating expense (including taxes) during the entire life of the new plant inclusive of cost of repairs and natural depreciation, but exclusive of functional depreciation,
e = same as E for the old property during its remaining life,
K and k = total equated or true annual cost during the entire life of the new property and the remaining life of the old property, respectively,
v = depreciated value of the old property,
U and u = unit cost of production of the new and of the old property, respectively,
Y and y = annual output in units of the new and of the old property, respectively,

Then
$$U = K/Y = [Cr + E + (C - S)F]/Y \tag{296}$$
$$u = k/y = [vr + e + (v - s)f]/y. \tag{297}$$

If the annual unit cost of production of the new property is the same as that of the old

$$U = u$$

and
$$[Cr + E + (C - S)F]/Y = [vr + e + (v - s)f]/y. \tag{298}$$

If the right-hand member of the equation is less than the left-hand member, it is cheaper to retain the old unit; if it is greater, then it is time to sell or scrap the old and buy the new.

For ordinary use, equation (298) may be reduced to a simpler form. Thus, if $U = u$, $Y = y$ and $F = f$, which is frequently the case, equation (298) may be reduced to the form

$$v = C + [(E - e) + f(S - s)] \div (r + f). \tag{299}$$

Furthermore, F is generally equal to f or nearly so, in which case equation (298) may be expressed

$$v = C + \frac{E - e}{r + f} = C - \frac{e - E}{R}. \tag{300}$$

When functional depreciation is non-existent, $f = 0$ and equation (299) becomes

$$v = C - (e - E)/r. \tag{301}$$

TABLE 124

ELECTRIC OPERATING EXPENSES — YEAR 1922

(Including Amortization and Depreciation)

Commonwealth Edison Company

	Total Expense, Thousands of Dollars	Per Cent of Total Expense	Cost per Kw.-hr., Mills
Steam Power Generation			
Superintendence	80.88	0.285	0.036
Boiler Labor	600 44	2.110	0.270
Engine Labor	242 82	0 853	0 109
Electrical Labor	211 13	0.743	0 095
Miscellaneous Labor	102 53	0.362	0.046
Fuel for Steam	12,175 13	42 800	5.470
Water for Steam	4.41	0.016	0 002
Lubricants	32 67	0.115	0.014
Power Plant Supplies and Expenses	115.56	0.407	0.052
Maintenance of Steam Power Buildings and Fixtures	123.61	0.435	0
Maintenance of Boilers and Boiler Plant Equipment	442 86	1.560	0 199
Maintenance of Steam Power Plant Piping	31.49	0.111	0 014
Maintenance of Steam Power Electric Generating Equipment	207 62	0 731	0.093
Maintenance of Steam Power Plant Switchboard and Wiring	40 70	0 143	0 018
Maintenance of Miscellaneous Steam Power Plant Equipment	53 50	0 189	0 024
Total Steam Power Generation	11,465 38	50 860	6 497
Electric Energy Purchased, Exchanged or Transferred			
Electric Energy Purchased	1,022 40	3.590	0 459
Total Electric Energy Purchased, etc.	1,022.40	3 590	0 459
Transmission			
Operation of Transmission Lines	119 39	0 420	0 054
Maintenance of Transmission Lines	74 92	0 263	0 033
	194 31	0.683	0 087
Storage Battery			
Storage Battery Labor	30.10	0.106	0 013
Storage Battery Supplies	1 19	0 004	0 000
Miscellaneous Storage Battery Expenses	25 79	0 091	0 012
Maintenance of Storage Battery Buildings	3 54	0 012	0 002
Maintenance of Batteries	35 86	0 127	0.016
Maintenance of Accessories	2 16	0 007	0 001
Total Storage Battery	98 64	0.347	0 044
Distribution			
Distribution Superintendence	199.17	0.700	0 089
Substation Wages	589 46	2 070	0.265
Substation Supplies and Expenses	148.47	0.526	0.066
Operation of Distribution Lines	368.90	1.296	0.166
Maps and Records	19 67	0.069	0.007
Inspecting and Testing Meters	270 86	0.952	0 122
Inspecting and Changing Transformers	60.17	0 211	0 027
Removing and Resetting Meters	121 15	0.427	0 054
Miscellaneous Distribution Operating Labor	54.61	0.192	0.024
Miscellaneous Distribution Supplies and Expenses	13 05	0.046	0.006
Maintenance of Substation Buildings	28.50	0 100	0.013
Maintenance of Substation Equipment	162 99	0.573	0.073
Maintenance of Overhead Distribution Pole Lines	105.30	0 370	0.047

TABLE 124 — *Concluded*

	Total Expense, Thousands of Dollars	Per Cent of Total Expense	Cost per Kw.-hr., Mills
Maintenance of Overhead Distribution Conductors.	25 32	0 089	0 011
Maintenance of Underground Distribution Conduits. .	17 33	0 060	0 008
Maintenance of Underground Distribution Conductors	77 01	0 271	0.035
Maintenance of Services	25 35	0 089	0.011
Maintenance of Meters	100 61	0 357	0.045
Maintenance of Transformers	47 92	0 167	0 021
Total Distribution .	2435 84	8 565	1 090
Utilization			
Municipal Street Incandescent Lamps	33 98	0 120	0 015
Commercial Arc Lamps Labor			
Commercial Arc Lamps Supplies and Expenses .	2 23	0 008	0 001
Commercial Incandescent Lamps	1109 87	3 910	0 498
Inspecting and Adjusting Customers' Installations	226 84	0 795	0 105
Leased Appliances Expense	52 73	0 185	0 024
Miscellaneous Utilization Expense	106 51	0 375	0 048
Maintenance of Commercial Incandescent Lamps and Equipment	40 21	0 141	0 018
Maintenance of Leased Appliances and Fixtures	82 05	0 288	0 037
Total Utilization	1654 12	5 822	0 746
Commercial			
Superintendence . .	46 39	0 163	0 021
Meter Reading - Salaries and Expense	272 10	0 956	0 122
Collecting — Salaries and Expense	237 71	0 837	0 107
Consumers' Accounts -- Salaries	914 55	3 218	0 411
Contract Department -- Salaries	9 27	0 036	0 004
Office Supplies and Miscellaneous Expense	122 75	0 430	0 055
Total Commercial	1602 77	5 640	0 720
New Business			
Superintendence.	35 21	0 124	0 016
Office Salaries. . . .	126 93	0 448	0 057
Soliciting .	107 21	0 377	0 048
Miscellaneous Supplies and Expenses	52 79	0 186	0 024
Advertising . . .	429 60	1 510	0 193
Wiring and Appliances	27 10	0 095	0 012
Total New Business	778 87	2 740	0 350
General and Miscellaneous			
Salaries and Expenses of General Officers	253.48	0 892	0 114
Salaries and Expenses of General Office Clerks	748 57	2 633	0 336
General Office Supplies and Expenses	241 51	0 850	0 109
Stationery and Printing	34 41	0 121	0.015
General Office Rents	551.90	1 942	0 248
Maintenance of General Buildings	105 11	0 369	0 047
General Law Expense .	189.82	0 668	0 085
Injuries and Damages	87.18	0 306	0 037
Insurance. . . .	161.17	0 567	0.072
Relief Department Pensions and Welfare Work	195 56	0 688	0.088
Miscellaneous General Expense	310 39	1 090	0.138
Depreciation. . .	3139 94	11 056	1.415
Amortization of Intangible Capital	147.28	0 518	0 066
Total General and Miscellaneous	6166 32	21 700	2 770
Grand Total Electric Operating Expenses	28,418.95	100 000	12.763
Total Kw.-hr. Generated and Purchased .	2,225,442,875		

If it is time to replace the old equipment with the new, it has reached the end of its economical life and its depreciated value is equal to its salvage value and $v = s$. If the new and old units have the same output $Y = y$. Assuming $Y = y$ and $v = s$, equation (298) reduces to the form

$$Cr + E + (C - S)F = rs + e. \tag{302}$$

This relationship is shown graphically in Fig. 678.

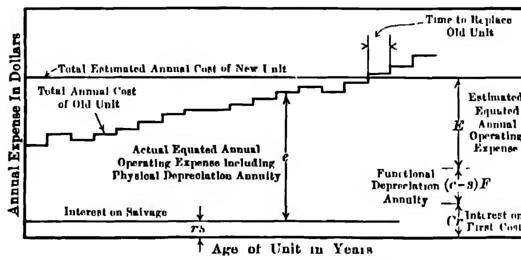


FIG. 678. Unit-cost Depreciation Method Showing Time of Replacement on Account of Obsolescence or Inadequacy.

If the annual operating expenses, O, other than depreciation, are the same for the new as for the old property, we have

$$E = D(C - S) + O$$
$$e = d(v - s) + O$$

in which D = depreciation annuity for the total life of the old property,
d = depreciation annuity for the remaining life of the new property.

Substituting these values for E and e in equation (302) and reducing, we have

$$v = S + (D + r)\ (C - S) \div (d + r) \tag{303}$$

which gives identically the same results as the ordinary sinking-fund formula, equation (294).

E and e, equated annual operating expenses, are calculated as follows: Take the total cost of labor, repairs, supplies and other items for the first year the property is in operation and find what this would amount to at compound interest for the years remaining in the estimated life of the plant. Similarly, the operating expense for each succeeding year is compounded for the respective remaining years. Add these various sums together and multiply by the annual deposit in a sinking fund which, if started at the beginning of the assumed life, will equal 1 at the expiration of the assumed economic life. The resulting product is the equated annual operating expense.

Example 92. — An engine, purchase price \$6000.00, is four years old and the yearly cost of operation, including supplies, taxes, insurance, labor and repairs (but not including functional depreciation) for these years is as follows: first year \$3000.00, second year \$3250.00, third year \$3750.00, fourth year \$4500.00. If the interest rate is 6 per cent, required the equated annual operating costs.

Solution. — \$3000.00 @ 6 per cent compounded annually for 3 yr. = $3000 \times (1.06)^3 =$	\$3573.05
\$3250.00 @ 6 per cent for 2 yr. = $3250 \times (1.06)^2 =$	3651.70
\$3750.00 @ 6 per cent for 1 yr. = $3750 \times 1.06 =$	3975.00
\$4500.00, no interest	4500.00
Total Cost	\$15,699.75

\$3590.21 annually placed at compound interest at 6 per cent for 4 yr. would amount to \$15,699.75; thus, from equation (292*d*),

$$15{,}699.75 \times 0.06 \div (\overline{1.06}^4 - 1) = \$3590.21.$$

This is the equated annual operating cost.

Example 93. — Suppose a new type of engine is on the market, suitable for the service performed by the one mentioned in Example 92, but more economical in operation. The new engine costs \$10,000, and its estimated functional life and salvage values are fifteen years and \$1000, respectively. Assuming that the salvage value of the old engine is \$400 and that the estimated equated annual operating cost of the new units is \$2600 per year, will it pay to junk the old one? Assume interest rate on the sinking fund deposits to be 6 per cent.

TABLE 125

TYPICAL WEEKLY POWER-PLANT REPORT

The Detroit Edison Company
Connors Creek Plant
For Week Ending September 1st, 1923

Lb. of coal per kw-hr. delivered	1 504
B.t.u. per kw-hr. delivered	20,048.0
Overall thermal efficiency, per cent	17.03
Plant water rate — lb. per kw-hr. delivered	13.45
Combined boiler furnace and grate efficiency, per cent	76.0
Condensing water inlet temperature, deg. fahr.	68.0
Output	
Net delivered, kw-hr.	12,446,600.0
Average output per day, kw-hr.	1,778,086.0
Ratio net delivered to total generated, per cent	97.21

Coal		
Coal consumed, lb.		18,725,100.0
Average coal consumed per day, tons		1,338.0
Analysis	Total moisture, per cent	3.995
	Ash (dry coal), per cent	10.0
	Heating value (as fired), B.t.u.	13,330.0
	Heating value (dry)	13,885.0
Refuse analysis		
Average carbon in refuse, per cent		15.15

Solution. — Here $C = 10{,}000$, $r = 0.06$, $E = 2600$, $S = 1000$, $s = 400$, $e = 3590.21$ (from example 92).

From equation (292d), $F = 0.06 \div (1.06^{15} - 1) = 0.0429$.

Substituting these values in equation (302) and solving

$$10{,}000 \times 0.06 + 2600 + (10{,}000 - 1000)\,0.0429 < 0.06 \times 400 + 3590.21$$
$$3586.1 < 3614.21$$

Since 3586.1 is less than 3614.21, it would pay to make the change.

Example 94. — A steam turbine unit, initial cost $25,000, has been in operation for ten years and its equated annual operating cost for this period is $10,000. The depreciation annuity for this equipment is based on an assumed functional life of fifteen years with interest at 6 per cent. A new and more economical unit of the same capacity as the old one can be purchased completely installed for $40,000. The estimated equated annual operating expense of the new unit is $8000. If the old unit can be sold for $1000 net, what is its depreciated value? Assume a functional life of 20 years for the new unit and a salvage value of $1500.

TABLE 126

OUTPUT AND COST DATA

California Edison Company

(Year 1922)

Transmitted from hydro-electric plants, kw-hr.		1,058,703,776
Transmitted from steam plants		72,718,357
Electric energy purchased from other sources		67,504,236
Total transmitted and purchased		1,198,926,369
Electric energy used in other departments	23,184,447	
Transmission loss	187,077,651	
Sub-station loss	717,378	
Unaccounted for	86,072,802	
Total		297,052,278
Total sold, light and power		901,874,091

Operating revenue:		
Sale of electricity	$15,758,686.60	
Rent of meters	345 75	
Joint electric	3,900 00	
Merchandise and jobbing	24,661 64	
Rent of appliances	9,160 03	
Total	$15,796,754 02	
Water dept.	42,823 44	
Total electric and water		$15,839,577.46
Operating expense:		
Production and transmission	$2,379,815.80	
Distribution	1,251,301 26	
Commercial	816,308 38	
General office	629,195 24	
Taxes	1,725,189 39	
Uncollectible bills	19,940 00	
Depreciation	1,850,190 17	
Water dept	44,835 60	
Total electric and water		8,717,105 84
Net operating revenue		7,122,471.62
Net non-operating revenue		1,142,648 29
Total net revenue		$8,265,119.91
Total investment	$96,691,000 00	
Cost of operation per kw-hr. transmitted and purchased, cents		0.722
Cost of operation per kw-hr. sold, cents		0.960

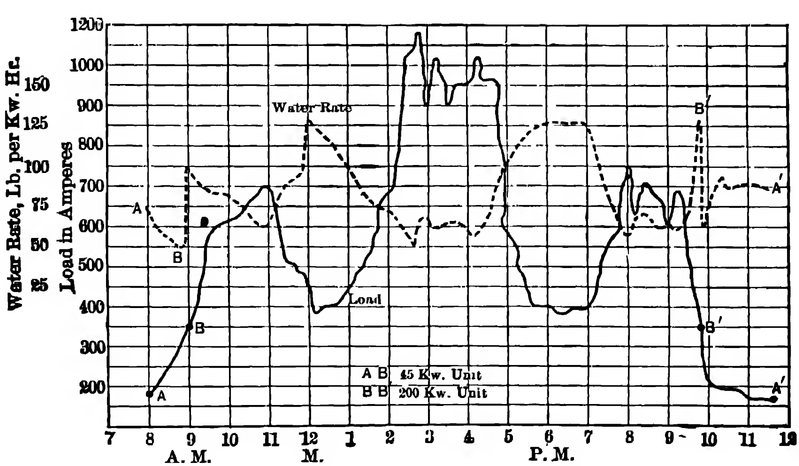


FIG. 679. Daily Load Curve, Showing Influence of Variable Generator Load on Steam Economy.

TABLE 127

POWER COST WORCESTER (MASS.) ELECTRIC LIGHT CO.

(1922)

Equipment: Boilers: 6 Stirling, 2 Edge Moor, 6 Bigelow-Hornsby.
Turbine Units: 2 20,000-kva., 2 7800-kva.
Stokers: Underfeed for bituminous, Forced-draft chain grate for anthracite.

Fuel: 66,956 tons of bituminous at \$7.52 and 13,613 tons of anthracite at \$5.12 per ton.

Output: 73,256,100 kw.-hr. Maximum load 32,600 kw., yearly load factor 0.256.

PRODUCTION EXPENSE

Superintendence	Total		Per Kw.-hr., Cents		Per Cent of Total	
Labor:		\$17,058		0 0233		2 13
Boiler	\$35,599		0 0486		4 44	
Turbine	18,905		0.0258		2 37	
Electrical	6,824		0 00933		0 86	
Miscellaneous	7,109		0 0097		0.88	
Total labor		68,437		0.09343		8 55
Supplies:						
Boiler fuel	\$580,583		0 793		72.40	
Water for steam	322		0 00044		0 04	
Lubricants	3,653		0 00499		0 47	
Station	29,770		0 0406		3 75	
Total supplies		614,328		0 83903		76.66
Maintenance:						
Station structures	\$21,237		0 0290		2 65	
Boiler plant equip.	31,059		0 0424		3 79	
Turbine units	21,217		0 02895		2 65	
Generator equip.	8,755		0 01185		1.09	
Accessory elec. equip.	1,475		0 002015		0 02	
Miscellaneous equip.	19,717		0 0269		2.46	
Total maintenance		103,460		0 14115		12 66
Grand Total		\$803,283		1 097		100 00

Solution. — Here $C = 40{,}000$, $r = 0.06$, $E = 8000$, $S = 1500$, $s = 1000$.

$$F = 0.06 \div (\overline{1.06}^{20} - 1) = 0.02718$$
$$f = 0.06 \div (\overline{1.06}^{15} - 1) = 0.04296.$$

Substituting these values in equation (298), noting that $Y = y$, we have

$$40{,}000 \times 0.06 + 8000 + (10{,}000 - 1500)\,0.02718 = 0.06\,v + 10{,}000 + (v - 1000)\,0.04296,$$

from which $v = \$6546.13$, the present or depreciated value of the old unit.

According to the straight-line law, equation (290) the present value is \$9333.33, and according to the sinking-fund law, equation (294), it is \$11,700.00. The depreciated value of any used device, or its true worth

to a purchaser, according to the unit-cost depreciation law, is dependent solely upon its equated annual operation costs when compared with those of a more economical device which can do the same work, and not upon its first cost or age.

Power Plant Accounts: W. A Miller, Power, Nov. 1, 1921, p. 680; Dec. 13, 1921, p. 934.

Improvements and Replacements Power Plant Engrg., Jan. 1, 1920, p. 63.

Fundamentals of Public Utility Rates. National Engr., April, 1924, p. 151.

365. Operating Costs. — General Division. — The distribution of the operating costs depends largely upon the size and nature of the plant. In the small isolated station the term "operating costs" without qualification refers to the generating or station operating costs, exclusive of fixed charges. These costs are commonly divided as follows:

1. Labor and attendance.
2. Fuel and water.
3. Oil, waste, and sundry supplies.
4. Repairs and maintenance.

In some of the larger isolated stations, a more extensive division is often made but there appears to be no accepted standard.

In large central stations, the operating costs are divided under the major headings of

1. Production expenses.
2. Transmission expenses.
3. Electric storage expenses.
4. Utilization expenses.
5. Commercial expenses.
6. New business expenses.
7. General and miscellaneous expenses.

The extent of the subdivisions under each subheading depend upon the size and nature of the plant. See Table 124.

A number of large central stations limit the major headings to

1. Generation.
2. Administration.
3. Distribution.

Some companies include all or part of the fixed charges under the major heading; others limit the operating costs to expense, which is dependent only on the output. Because of this diversity in bookkeeping, comparisons of the cost of power based on the annual reports are without purpose. A few annual reports illustrating the different systems of accounting are reproduced in the accompanying tables.

366. Labor, Attendance, Wages. — The minimum number of men required to handle a given plant is approximately a fixed quantity and it is seldom possible to so arrange the work that any material reduction can be effected. Until very recently it has been the universal custom to pay wages on a "flat rate" basis; that is, the attendant is given a fixed sum per day or month irrespective of the amount of work required or the economy of operation. In some cases, however, the bonus system has been successfully adopted. For example, in the hand-fired boiler room

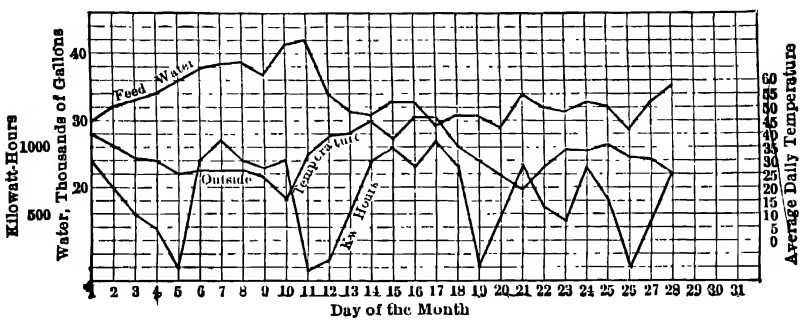


FIG. 680. Monthly Load Curve of Combined Heat and Power Plant, Armour Institute of Technology.

the coal consumption is determined for a given period of time with ordinary careful firing, and the fireman is offered a reasonable percentage on the saving of coal which he is able to effect over this record by special care and attention to the keeping of fires always in the best condition, avoiding the blowing off of steam, using as little coal as needed for banking fires, and in other ways. Where careful records are kept of supplies, repairs, and renewals, the bonus is also applicable to electricians, oilers, and other employees.

Labor should always be estimated or recorded as so many dollars per month or per year and not merely in terms of the output unless the load factor is definitely known; otherwise comparisons are misleading. For example, consider two plants of 500 kw. capacity, each with labor charges, say, of \$400 per month. Suppose the output of one is 100,000 kw-hr. per month and that of the other 40,000 kw-hr. per month. The monthly charges are evidently the same, viz., \$400, but the cost per kw-hr. differs widely, being 0.4 cent in the first case and 1 cent in the latter.

The cost of labor varies so much with the location of the plant and the conditions of operation that general figures are of little value except as a rough guide. Specific figures will be found in the accompanying tables.

For a summary of labor costs in large central stations see "Central-Station Labor Costs," Electrical World, Nov. 16, 1912, p. 1031.

TABLE 128

COMPARISON OF PRODUCTION EXPENSES PER KILOWATT-HOUR OF OUTPUT FOR CONNORS CREEK POWER HOUSE OF DETROIT EDISON COMPANY

	Twelve-Month Periods Ending							
	1919		1920		1921		1922	
	June 30	Dec 31	June 30	Dec 31	June 30	Dec 31	June 30	Dec. 31
	Cents	Cents	Cents	Cents	Cents	Cents	Cents	Cents
Operation:								
Superintendence	0.010	0.010	0.010	0.012	0.015	0.011	0.013	0.012
Wages	0.062	0.060	0.062	0.076	0.079	0.069	0.065	0.058
Fuel	0.394	0.407	0.471	0.652	0.682	0.532	0.438	0.448
Water								
Lubricants	0.002	0.002	0.001	0.002	0.002	0.002	0.001	0.001
Station supplies and expense	0.010	0.007	0.011	0.012	0.010	0.010	0.010	0.007
Total	0.478	0.486	0.555	0.754	0.788	0.627	0.527	0.526
Maintenance								
Station buildings	0.008	0.011	0.010	0.011	0.012	0.011	0.012	0.011
Steam equipment	0.024	0.029	0.036	0.038	0.043	0.045	0.041	0.037
Electrical equipment	0.001	0.001	0.001	0.003	0.003	0.002	0.002	0.004
Total	0.033	0.041	0.047	0.052	0.058	0.058	0.055	0.051
Total production exp	0.511	0.527	0.602	0.806	0.846	0.685	0.582	0.577
Output (millions of kw. hr.)	383.25	445.54	488.06	479.43	485.19	527.12	555.90	613.26
Maximum demand in kw. (30 minute)	82,000	104,000	104,000	100,000	107,500	118,000	120,000	144,500
Average load (kw.)	43,700	50,800	55,600	54,600	55,300	60,200	63,500	70,000
Load factor	0.533	0.488	0.534	0.546	0.515	0.510	0.529	0.485
Coal per kw. hr. (lb.)	1.67	1.73	1.83	1.92	1.78	1.62	1.55	1.57
B.t.u. per kw. hr.	21,200	21,800	22,800	23,300	21,800	20,250	19,700	19,660

367. Cost of Fuel. — Tables 121 to 131 give specific examples of the cost of fuel in different sizes and types of steam power plants. It will be noted that this item varies considerably even with plants of the same general class. So much depends upon the grade and market price of the fuel, type and size of plant, and conditions of operation that no single item can afford a means of comparing fuel costs in different plants. Such items as "lb. coal per kw-hr.," "cost of fuel per kw-hr.," or the equivalent have their value in any accounting system, but fail utterly as a measure of the economy of operation unless accompanied by a statement of the qualifying conditions. For example, an inefficiently operated plant using a high-grade fuel may show a lower fuel consumption, lb. per kw-hr., than an economical plant using a low-grade fuel, and an uneconomical plant using a very cheap fuel may show a lower "cost of fuel per kw-hr."

than an efficiently operated plant using costly fuel. Similarly, two plants of the same size and type, and using the same fuel, may show considerable difference in both "lb. of fuel per kw-hr." and "cost of fuel per kw-hr." because of difference in load factor, even though both plants are efficiently operated for the given conditions. In a number of recent installations, the station operating records include the heat supplied by the fuel per kw-hr. generated ("B.t.u. per kw-hr.") and the cost of the fuel on a

TABLE 129

FUEL CONSUMPTION AND COSTS

Massachusetts Central Stations

(Year Ending 1922)

Station	Rated Boiler Hp. (Thousands)	Rated Kva (Thousands)	Yearly Output Kw-hr. (Millions)	Load Factor	Tons of Coal per Year (Thousands)	Cost of Coal per Ton (Dollars)	Lb. of Coal per Kw.-hr.
Cambridge	4.40	25.68[1]	38,434	. .	41.32	8.744	2.147
Edison, Boston	48.59	203.00[2]	419,816	0.295	358.66	6.945	1.708
Edison, Brockton	5.70	23.15	43,197	0.267	41.62	7.600	1.926
Fall River	5.20	17.81	39,717	0.318	193.78[3]	1.351[4]	1.640[5]
Haverhill	3.20	16.37	14,879	0.130	16.53	7.705	2.222
Lowell	4.88	28.40	40,698	0.205	36.77	7.721	1.805
Lynn	4.78	20.12	22,042	0.169	22.53	6.720	2.041
Malden	1.20	34.00	1,082	0.455	2.21	9.104	.....
Malden	1.20	. . .	. . .	. . .	297.92[3]	1.650[4]	.....
New Bedford	15.60	90.00	119,257	0.189	130.95	6.288	2.195
Salem	4.50	28.12	65,741	0.334	55.44	6.573	1.688
Worcester	12.55	55.60	73,256	0.188	66.95	7.752	.....
Worcester	....	. . .	. .	. . .	13.61[6]	4.520	.....

[1] 6500 hp. engines in addition. [2] Kw. Boilers not installed for last 30,000 kw. unit.
[3] Oil, thousands of bbls. [4] Per bbl. of oil.
[5] Lb. oil per kw.-hr. [6] Screenings.

heat basis (cents per 10,000 B.t.u. or B.t.u. for 1 cent). These two items in connection with the load factor offer a satisfactory criterion of the fuel economy for plants of the same general design. Large central stations, with individual units of 20,000 to 60,000 kw. rated capacity and a yearly load factor of 50 per cent or more, have been credited with a yearly performance of 18,000 B.t.u. per kw-hr. generated, corresponding to an overall thermal efficiency of approximately 19 per cent. With 11,000 B.t.u. screenings, this is equivalent to approximately 1.6 lb. coal per kw-hr. and with 13,000 B.t.u. coal, about 1.4 lb. coal per kw-hr. Better results than this have been obtained for brief periods of operation, but when averaged over a considerable period of time, the standby losses, such as

coal burned in banking fires, heat lost in blowing down boilers, lower efficiency in operating at underloads and overloads, and the like, reduce the overall efficiency to substantially that given above. The coal consumption per kw-hr. for a number of medium-sized central stations in Massachusetts for the year 1922 is given in Table 129.

In estimating the cost of fuel for a proposed installation the logical procedure is as follows:

1. Construct load curves for the probable power requirements.
2. Calculate the total weight of steam supplied from the load curve.
3. Transfer the total steam requirements to the unit water-rate basis.
4. Reduce the average unit water rate to "B.t.u. supplied by the steam per unit output."
5. Divide the average B.t.u. supplied by the steam per unit output by the estimated overall boiler efficiency, considering all standby loss. This gives the B.t.u. supplied by the fuel per unit output.
6. Reduce the cost of fuel to "cost per 10,000 B.t.u.," or "B.t.u. per 1 cent."
7. Multiply item 5 by item 6 and divide by 10,000, if the 10,000 B.t.u. basis is used, and divide item 5 by item 6 if the "B.t.u. per cent" basis is used. This gives the average cost of fuel per unit output for the required period.

The construction of the load curves is the most important item, since the cost of the fuel per unit output is primarily a function of the load factor.

The total weight of steam is calculated from the load curve by considering the unit water rate of the prime mover and steam-driven auxiliaries at the variable loads, and the time element.

The heat supplied by the steam is measured above the temperature of the feedwater. In plants where exhaust is used for heating or manufacturing purposes, only the difference between the heat supplied to the prime movers and steam-driven auxiliaries and that of the exhaust utilized for heating is charged to power.

Current practice gives an average efficiency (based on yearly operation) of boiler and furnace of 75–80 per cent for large lighting and power stations with yearly load factor of 0.45 or more, and 65–75 per cent for similar stations with load factor between 0.35 and 0.40. For very low load factors, 0.25 and under (as in connection with large manufacturing plants, tall office buildings, and other plants operating on a twelve-hour basis), this overall efficiency seldom exceeds 60 per cent. With these figures as a guide, the cost of fuel per unit output may be roughly approximated.

TABLE 130

COST OF LABOR, SUPPLIES AND MAINTENANCE

Massachusetts Central Stations

Year Ending 1922

	Cambridge	Boston Edison	Brockton Edison	Fall River	Haverhill	Lowell	Lynn	New Bedford	Salem	Worcester
Rated capacity, thousand kva.	25.68	203.00†	23.15	17.81	16.37	28.40	20.12	90.00	28.12	55.60
Output, million kw-hr.	38.43	419.81	43.19	39.71	14.88	40.69	22.04	119.25	65.74	73.25
Load factor*	.....	0.295	0.267	0.318	0.136	0.205	0.169	0.189	0.334	0.188
	Cost in Mills per Kilowatt-hour									
Operation:										
Superintendence	0.122	0.080	0.097	0.138	0.335	0.242	0.176	0.141	0.237	0.233
Engine labor	0.386	0.305	0.412	0.392	0.774	0.448	0.969	0.227	0.298	0.258
Boiler labor	1.010	0.411	0.464	0.380	1.265	0.750	1.326	0.286	0.488	0.486
Elec. labor	0.324	0.189	0.264	0.120	0.456	0.275	0.356	0.072	0.186	0.093
Misc. labor	0.463	0.091	0.103	0.102	0.216	0.173	0.298	0.093	0.054	0.097
Boiler fuel	9.400	5.935	7.560	6.580	8.460	7.260	11.971	5.481	5.520	7.930
Water for steam	0.198	0.096	0.004	0.288	0.075	0.007	0.654	0.097	0.331	0.004
Lubricants	0.056	0.033	0.027	0.043	0.064	0.024	0.072	0.039	0.039	0.050
Station supplies	0.071	0.229	0.111	0.125	0.500	0.146	0.374	0.058	0.194	0.406
Total	12.030	7.369	9.042	8.168	12.145	9.325	16.196	6.494	7.347	9.557
Maintenance:										
Station structures	0.381	0.065	0.086	0.050	0.410	0.319	0.786	0.044	0.060	0.290
Boiler plant equipment	0.422	0.412	0.426	0.858	0.484	0.364	0.768	0.258	0.664	0.417
Steam engines	0.165	0.014	....	....	0.003	..	0.236	0.188	.....	....
Turbine units	0.119	0.156	0.047	0.512	0.254	0.111	0.339	0.222	0.351	0.290
Elec. generator equipment	0.037	0.042	0.015	0.044	0.005	0.036	0.074	0.014	0.090	0.120
Accessory elec. equipment	0.068	0.037	0.046	0.048	0.051	0.107	0.187	0.030	......	0.020
Misc. power-plant equipment	.....	0.001	0.030	.....	0.277	0.135	0.212	0.044	0.174	0.269
Total	1.192	0.727	0.650	1.512	1.484	1.072	2.602	0.800	1.339	1.406
Grand Total	13.222	8.096	9.692	9.680	13.629	10.397	18.798	7.294	8.686	10.963

* Kw.-hr. ÷ (8760 × rated kva. × 0.8) † Kw.

368. Oil, Waste, and Supplies. — These items approximate from a fraction to 5 per cent of the total operating expenses. Tables 124, 127, 128 and 130 give some idea of current practice in different classes of power plants.

369. Repairs and Maintenance. — This item ordinarily refers to the cost of keeping the plant in running order over and above the cost of labor or attendance, and depends upon the age and condition of the plant and the efficiency of the employees. Tables 124 to 133 give the cost of repairs and maintenance for a wide range in power-plant practice for the years 1920–23.

370. Cost of Power. — The actual cost of producing power depends upon the geographical location of the plant, the size of apparatus, the design, conditions of loading, system of distribution, and the method of accounting. Tables 124 to 133 compiled from various sources give the detailed costs of a large number of central and isolated stations.

1921 Experience Exchange. National Assoc. of Building Owners and Managers, Edison Bldg., Chicago.

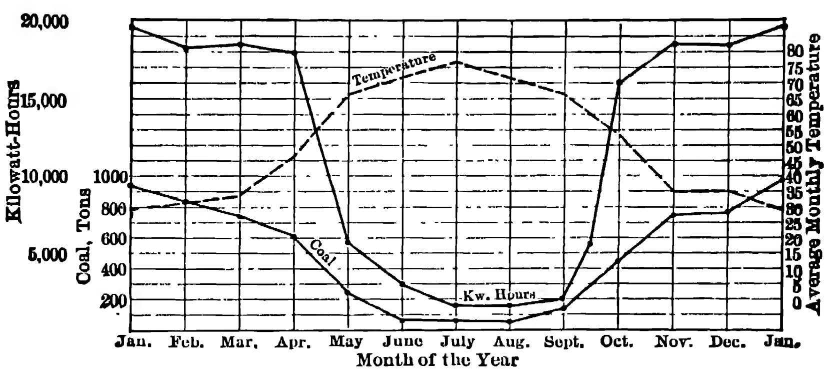


Fig. 681. Yearly Load Curve, Showing Influence of Temperature on Coal Consumption. Combined Heat and Power Plant, Armour Institute of Technology.

371. Elements of Power-plant Design. — The real problem which confronts the designing engineer is not so much the selection and arrangement of apparatus for a given set of conditions as it is to foresee the conditions under which the plant is likely to operate. For this reason, the plans for the station should be examined and approved by the owners and expert service employed at the outset. It is not sufficient to have a mechanically perfect plant, though of course proper installation is of prime importance. The choice of fuel, selection of type of prime mover, size of units, provision for future expansion, method of establishing the station heat balance, and similar factors have considerable weight in the

TABLE 131

Investment and Operating Cost of a 200-kw. Central Station

Load Factor 50 Per Cent

Steam Pressure 175 Lb. for All Units Except Corliss Engines
Steam Pressure 150 Lb. for Corliss Engines
Cost of 13,500 B.T.U. Coal $7 00 Per Net Ton Delivered

A — Steam Equipment Designed for Saturated Steam

Non-Condensing Steam Prime Movers
Back Pressure Atmospheric

Condensing Steam Prime Movers
Engines condensing to 26" Vacuum
Turbines condensing to 28" Vacuum

N.E.L.A. (T5-21)

Investment	2-200 Kw. Uni-flow Engine	3-100 Kw Uni-flow Engine	2-200 Kw. Counter flow Engine	3-100 Kw. Counter flow Engine	2-200 Kw. Turbines	3-100 Kw Turbines	2-200 Kw Corliss Engine Belted	3-100 Kw. Corliss Engine Belted	2-260 Kw. Uni-flow Engines	3-100 Kw Uni-flow Engines	2-200 Kw. Counter flow Engines	3-100 Kw. Counter flow Engines	2-200 Kw Turbines	3-100 Kw Turbines	2-200 Kw. Corliss Engines Belted	3-100 Kw. Corliss Engines Belted
Real estate	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500
Brick building	35,200	35,800	35,200	35,800	26,000	28,000	43,000	45,000	38,000	38,600	38,000	38,600	28,800	30,800	45,800	47,800
Generating units, del. and erected	31,954	32,250	25,504	26,634	16,750	19,449	23,380	24,450	31,954	32,250	25,504	26.634	19,460	23,100	23,380	24,450
Switchboards and street ltg. transformers	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000
Electric wiring and ducts	3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000	3.500	4.000	3,500	4,000	3,500	4,000	3.500	4,000
Piping complete	6,500	6,500	6,500	6,500	6,500	6,500	6,500	6,500	9,500	10.500	9,500	10,500	9,500	10,500	9,500	10,500
Condensing equipment									7,000	10,500	7,000	10,500	7,000	10,500	7,000	10,500
Foundations, exclusive building	4.200	3,900	4,200	3,900	2,500	2,800	4,500	4,200	4,700	4,400	4,700	4,400	2,800	3,100	5,000	4,700
Oil filters and tanks	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800
Railroad siding	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3.200	3,200	3,200	3.200	3,200	3,200
Boilers, delivered and bricked in	13,000	13.000	13,000	13,000	15.400	15,400	15,400	15,400	13,000	13,000	13.000	13.000	13,000	13,000	15,400	15,400
Feedwater heater	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900
Feed Pumps	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500
Steel stack and flues	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800
Total $	112,554	114,150	106,104	108,534	88,850	94,849	114,480	118,250	125,854	131,950	119,404	126,334	102,260	113,700	127,780	136,050
Cost of operation, fixed charges 15%	16,883	17,123	15,915	16,280	13,328	14,227	17,172	17,737	18,878	19,793	17,910	18,950	15,339	17,055	19,167	20,407
Fuel	13,310	13,302	16,290	15,692	20,974	20,902	18,100	17,602	11,650	11,900	15,500	15,100	13,100	13,600	17,200	16,700
Labor and supt.	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8.060	8,060	8,060	8,060
Lubricants	400	400	400	400	200	200	400	400	400	400	400	400	200	200	400	400
Miscellaneous supplies	350	350	350	350	300	300	350	350	350	350	350	350	300	300	350	350
Repairs	1,858	1,870	1,600	1,645	1,009	1,090	1,611	1,654	2,138	2,290	1,882	2,065	1,229	1,443	1,891	2,074
Total $	40,861	41,105	42,615	42,427	43,871	44,779	45,693	45,803	41,476	42,793	44,102	44,925	38,228	40,658	47,068	47,991

TABLE 131 — *Continued*

Cost per kw-hr. operation (cents)	2.74	2.74	3.05	2.98	3.48	3.49	3.26	3.21	2.58	2.62	2.99	2.96	2.62	2.70	3.19	3.15
Cost per kw-hr. fixed charges	1.93	1.95	1.82	1.86	1.52	1.62	1.96	2.03	2.15	2.26	2.05	2.16	1.75	1.94	2.19	2.33
Cost per kw-hr. total	4.67	4.69	4.87	4.84	5.00	5.11	5.22	5.24	4.73	4.88	5.04	5.12	4.37	4.64	5.38	5.48
Cost per kw peak	562.77	570.75	530.52	542.67	444.25	474.25	572.40	591.25	629.27	659.75	597.02	631.67	511.30	568.50	638.90	680.25
B — Steam Equipment Designed for Steam Superheated 100° Fahr																
Real estate	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500	2,500
Brick building	35,200	35,800	35,200	35,800	26,000	28,000	43,000	45,000	38,000	38,600	38,000	38,600	28,800	30,800	45,800	47,800
Generating units, del. and erected	31,954	32,250	25,504	26,634	16,750	19,449	23,380	24,450	31,954	32,250	25,504	26,634	19,460	23,100	23,380	24,450
Switchboards and street ltg transformers	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000	6,500	7,000
Electric wiring and ducts	3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000	3,500	4,000
Piping complete	7,000	7,000	7,000	7,000	7,000	7,000	7,000	7,000	10,000	11,000	10,000	11,000	10,000	11,000	10,000	11,000
Condensing equipment									7,000	10,500	7,000	10,500	7,000	10,500	7,000	10,500
Foundations, exclusive building	4,200	3,900	4,200	3,900	2,500	2,800	4,500	4,200	4,700	4,400	4,700	4,400	2,800	3,100	5,000	4,700
Oil filters and tanks	1,800	1,800	1,800	1,800	1,500	1,800	1,500	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800
Railroad siding	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200	3,200
Boilers, delivered and bricked in	15,100	15,100	15,100	15,100	18,200	18,200	18,200	18,200	15,100	15,100	15,100	15,100	15,100	15,100	18,200	18,200
Feedwater heater	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900	900
Feed pumps	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500	1,500
Steel stack and flues	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800	1,800
Total $	115,154	116,750	108,704	111,134	92,150	98,149	117,780	121,550	128,454	134,550	122,004	128,934	104,860	116,300	131,080	139,350
Cost of operation, fixed charges 15%	17,273	17,513	16,306	16,670	13,822	14,722	17,667	18,232	19,268	20,183	18,300	19,340	15,729	17,445	19,662	20,902
Fuel	12,725	12,660	15,400	14,820	20,300	19,900	17,860	17,300	11,100	11,500	14,600	14,700	13,150	14,000	17,350	16,700
Labor and supt.	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060	8,060
Lubricants	400	400	400	400	200	200	400	400	400	400	400	400	200	200	400	400
Miscellaneous supplies	350	350	350	350	300	300	350	350	350	350	350	350	300	300	350	350
Repairs	1,942	1,954	1,684	1,729	1,093	1,174	1,723	1,766	2,222	2,374	1,964	2,149	1,292	1,506	2,003	2,186
Total $	40,750	40,937	42,290	42,029	43,775	44,356	46,060	46,108	41,400	42,867	43,674	44,999	38,731	41,511	47,825	48,598
Cost per kw-hr. operation (cents)	2.68	2.68	2.96	2.89	3.42	3.38	3.24	3.18	2.52	2.59	2.89	2.93	2.63	2.75	3.22	3.15
Cost per kw-hr. fixed charges	1.97	2.00	1.86	1.90	1.58	1.68	2.02	2.08	2.20	2.30	2.09	2.21	1.79	1.99	2.24	2.39
Cost per kw-hr total	4.65	4.68	4.82	4.79	5.00	5.06	5.26	5.26	4.72	4.89	4.98	5.14	4.42	4.74	5.46	5.54
Cost per kw. peak	575.77	583.75	543.52	555.67	460.75	490.74	588.90	607.75	642.27	672.75	610.02	644.67	524.30	581.50	655.40	696.75

economy of operation. Each proposed installation is likely to be a problem in itself, and though similar plants may be used as patterns, each case should be worked out on its own merits.

In the case of an electric generating station, the first item to be considered is the kind of current, voltage, and distributing system best suited for the particular service to be rendered. In the older office buildings, stores, hotels, and small industrial plants having their own isolated plant, the direct-current, low-voltage (125–250), three-wire system predominates, but in many recent installations alternating-current generators are employed, so that minimum changes would be necessary in case central-station power is purchased.

In large industrial plants and central stations, alternating currents are the more common, the frequency and voltage depending upon the class of service. As a general rule, the smaller stations generate lower voltages than the larger ones. In the more recent designs, three-phase current generation predominates, not only because the energy can be transmitted very economically but also because the three-phase can be readily transformed into single or two-phase currents. The frequencies most generally employed are 25- and 60-cycle, with an increasing leaning toward the latter. The maximum voltage for which generators are designed is 14,000, but much higher values are used for transmission. The problem of selecting the voltage, kind of current, etc., for the particular service required should be left to the designing engineer at the outset.

One of the most important factors in the design of a power station is the determination of the probable load curve. This refers not only to the average yearly load but also to the maximum daily load which is likely to occur, the minimum daily load, temporary peak loads, and probable future increase. The load factor, which has such a marked bearing on the cost of operation, may be closely approximated from the daily load curves. Steam requirements for heating and industrial purposes, water supply, and other forms of energy requirements should be considered simultaneously with the electrical demands, since these factors largely influence the choice of prime mover. The curves in Figs. 671 and 673 are taken from the daily records of large power stations in Chicago and serve to illustrate the great variation in the electrical power demands for different days in the year. It is quite evident that an equipment based solely upon the average yearly requirements may not be adapted to the best economical operation.

The load curves for manufacturing plants may be predetermined with a fair degree of accuracy, since the power demands for various purposes may be readily segregated and analyzed, but with public utility concerns and certain classes of isolated stations the problem is largely a matter of

judgment. Thus, in the case of an industrial plant, the power requirements for lighting, manufacturing purposes, heating, ventilation, and sanitation may be closely approximated, since the size of building, exposure, number of floors, and the number of elevators afford a definite basis for analysis; but, with public utility concerns, the probable load depends largely upon the business acumen of the management in securing customers, the location of the plant, and future demands. In the latter case, the load curve is based chiefly upon the experience of similar plants under comparable conditions of operation.

TABLE 132

OPERATING COSTS OF TYPICAL INDUSTRIAL PLANTS

(1922)

(Harry Himelblau)

Plants	*A*	*B*
Number of boilers in plant	6	3
Rated capacity of boilers	1,400	500
Number of boilers in operation	4	3
Number of boilers in reserve	2	0
Boiler hp. in operation	900	500
Boiler hp. generated	1,200	500
Per cent rating of boilers in operation	133	100
Pressure lb. gage	150	100
Steam generated per hour, lb.	41,250	17,500
Steam generated per year, lb.	217,000,000	54,500,000
Fuel per year, tons	19,000	7,200
Average annual evaporation, lb. per lb. of coal	5.75	3.78
Average coal cost per ton	$6.89	$7.45
Annual operating costs		
Coal and handling	$134,000	$55,000
Engine and boiler-room labor	11,800	10,300
Repairs	7,400	1,000
City water	750	185
Boiler compounds	750	400
Oil and waste	2,300	300
Purchased power	30,000	20,000
Total	$187,000	$87,185
Cost per thousand pounds of steam (excluding cost of power)	$0.72	$1.23
Cost per kw-hr.		
Purchased power	0.02	0.02
Generated power	0.035	0.035

In any case, the greatest care should be exercised in estimating the maximum peak load which is likely to occur. High peak loads with low daily average necessitate the installation of large machines which are idle or operate uneconomically the greater part of the time and result in heavy fixed charges. The financial failure of many electric light and power plants is directly traceable to failure to consider the influence of maximum

peak loads on the ultimate cost of operation. In connection with central-station service, every customer represents a certain investment, regardless of the amount of power used. Even should he consume no power, his account would have to be carried on the books and a certain amount of equipment would have to be held in readiness to serve him. In order that every customer shall incur his share of the expense, the expense of the plant must be apportioned between the capacity and output costs. The heavier the peak load, the greater will be this charge, and, as in case with many small lighting plants where current is used but three or four hours a day, the "readiness to serve" charge becomes excessive, and either the station must operate at a loss or the unit cost will appear to be prohibitive.

The curves in Fig. 679 are taken from recording ammeter and recording steam-meter readings of a 200-kw. direct-connected and a 45-kw. belted generator set installed at the power plant of the Armour Institute of Technology, and serve to illustrate the influence of load on economy for very unfavorable conditions. At 8:00 A.M. the small unit is started up with initial load of about 150 amperes. As the load increases the water rate decreases, as is shown by the curve AB. At 9:00 A.M. the load is beyond the capacity of the small machine and the large unit is put into service. The increased water rate of the large unit over the requirements of the smaller is apparent from the sudden rise in the water-rate curve. This is due to the fact that the large unit is operating at only 20 per cent of its rating, against full load for the small one. The fluctuation of the water rate with the load variation is clearly shown. Evidently the two units are not of the proper size for the particular load conditions. During the heating months when live steam is necessary for "make-up" purposes, the unfavorable engine load has little effect on the ultimate economy, but during the summer months the loss from this cause may be a serious one.

The curves in Figs. 680 to 681 show that during the winter months, in a combined heating and power plant, the fuel requirements may be practically uninfluenced by the electrical demands, and an increase in electrical output does not effect an appreciable increase in fuel consumption; but the influence of the outside temperature is clearly indicated.

Steam Power Station Design: F. S. Clark, Power, May 24, 1921, p. 827.

372. Refinement in Power-Plant Design. — There is no question but that the installation of various appliances for utilizing the so-called waste-heat losses and the use of high-pressure and high-temperature steam, operating in heat cycles other than the Rankine, will result in increased overall thermal efficiencies far above those obtained without these refine-

TABLE 133

ANNUAL COST OF STEAM HEATING

Chicago Buildings
(1922)

Type of Building	O	O	O	O	D	H	H
No. of floors	19	20	21	19	11	21	17
Building vol., million cu ft.	7.5	13	9.5	3.7	11.3	7	18.1
Total steam, million lb	30.9	71.0	37.9	18.1	22.4	53.1	12.5
Total coal, one thousand tons	2.87	6.48	3.46	1.69	2.11	4.54	1.15
Coal delivered, dollars per ton	5.88	5.79	6.37	6.58	6.45	5.95	5.86
Boiler hp.-hr., millions	1.03	2.37	1.19	0.60	0.75	1.77	0.42
Lb steam per lb of coal	5.4	5.47	5.48	5.33	5.30	5.84	5.43
	Cost per 1000 Lb. of Steam, Cents						
Coal	54.6	53.2	61.5	61.7	60.9	51.0	53.9
Labor	19.3	20.5	21.4	24.6	22.7	16.8	17.3
Ash removal	1.9	1.8	2.9	2.8	2.4	2.2	2.3
Supplies	3.7	3.2	4.1	2.8	3.3	3.9	3.3
Repairs	11.6	9.7	14.0	7.5	12.1	13.8	15.2
Fixed charges	18.1	20.6	18.6	17.4	17.7	19.9	18.3
Total	$1.092	$1.09	$1.225	$1.168	$1.191	$1.076	$1.103

O, Office Buildings, D, Department Store, H, Hotels

ments. This is also true for plants operating on the **Benson super-pressure** or the **Emmett mercury-steam** principle; but just where the added heat economy will be balanced by increased first cost, complexity, and reliability of operation can only be determined by a careful study of all of the factors entering into the problem of power generation. Thermal gains can be calculated with a fair degree of accuracy and first cost can be estimated within reasonable limits, but reliability of operation can be judged only from actual experience. That rapid progress has been made is evidenced by the almost revolutionary ideas incorporated in the latest projects. Many engineers are very conservative and are slow to adopt any marked departure from established practice. This reluctance is not due to opposition but rather to an attitude of "waiting to be shown," with a general readiness to accept the innovations as soon as their value is definitely established. Fuel and labor costs and the load factor are the predominating influences in determining how far it is commercially feasible to carry out the thermal savings. Solely because of the steadily increasing cost of fuel, small non-condensing steam-electric plants, which do not permit the utilization of a large part of the exhaust for heating or process work, may soon be a matter of history. Electricity for this class of service will, in all probability, be furnished either by large central stations or some other type of prime mover having lower fuel costs. Even in the large steam-electric

TABLE 134

COST OF OPERATION, TALL OFFICE BUILDING *

A. Cost of steam.		Total steam production...	25,700,000 lb.
1. Coal...	$ 8,750.00	Cost per thousand lb...	60c
2. Ash removal	70.00	Steam to Generators...	22,500,000 lb.
3. Boiler-room labor....	3,000.00	Live steam for heating.	3,200,000 lb.
4. Boiler-room supplies and repairs....	775.00		25,700,000 lb.
5. Boiler-room depreciation	1,550.00	Total steam for heating..	16,000,000 lb.
6. All plant overhead	775.00	Live steam for heating	3,200,000 lb.
7. Rental value of space...	500.00		
	$15,420.00	Exhaust used in heating..	12,800,000 lb.

B. Cost of electricity.

1. 22,500,000 lb. steam @ 60c	$13,500.00
2. Labor for generators	3,000.00
3. Supplies and repairs for generators	1,500.00
4. Depreciation on generators	960.00
	$18,960.00
5. Credit exhaust for heating 12,800,000 lb. @ 60c.	7,680.00
	$11,280.00

Total output, 500,000 kw-hr.

Cost per kw-hr. = 2-1/4c.

Charge electric light sold	150,000 kw-hr. @ 2-1/4c.	$3,375.00
Charge electric system for..	60,000 kw-hr @ 2-1/4c	1,350.00
Charge elevators for	175,000 kw-hr. @ 2-1/4c.	3,945.00
Charge to general expense.	115,000 kw-hr. @ 2-1/4c.	2,610.00
		$11,280.00

Cost of heating and hot water.

1. Live steam, 3,200,000 lb. @ 60c	$ 1,920.00
2. Exhaust for heating 12,800,000 lb. @ 60c.	7,680.00
Total cost of heating.	$ 9,600.00

From these figures can be made up the

Power Plant Statement

			Debit	*Credit*
A.	1	Coal	$ 9,250.00	
	2	Ash removal	70.00	
	3	Boiler-room labor	3,000.00	
	4	Boiler-room supplies and repairs	775.00	
	5	Boiler-room depreciation	1,550.00	
	6	All plant overhead	775.00	
B.	2	Labor for generators	3,000.00	
	3	Supplies and repairs for generators	1,500 00	
	4	Depreciation on generators	960.00	
C.	1	Labor for heating system	450.00	
	2	Labor for electric system	550.00	
	3	Labor for elevators	750.00	
A.	4	Electric system		$ 1,900.00
A.	5	Heating system		10,050.00
A.	7	Elevators		4,695.00
A.	8	General expense		2,610.00
H.	1	Electricity sold		3,375.00
		Total or Account A9.0	$22,630.00	$22,630.00

* Standard form as recommended by the National Association of Building Owners and Managers, Executive Office, Edison Building, Chicago, Ill.

industrial plant with heavy electrical requirements and low demands for steam, it may be more economical to purchase central-station service than to generate current within the plant itself.

The curves in Fig. 682, by Hirshfeld and Ellenwood[1] are of interest in showing what may be expected in the way of coal consumption for a 200,000-kw. steam-turbine plant operating on various cycles and steam pressures. These curves are based on a capacity factor of 100 per cent, combined boiler efficiency 84 per cent, throttle temperature 700 deg. fahr.,

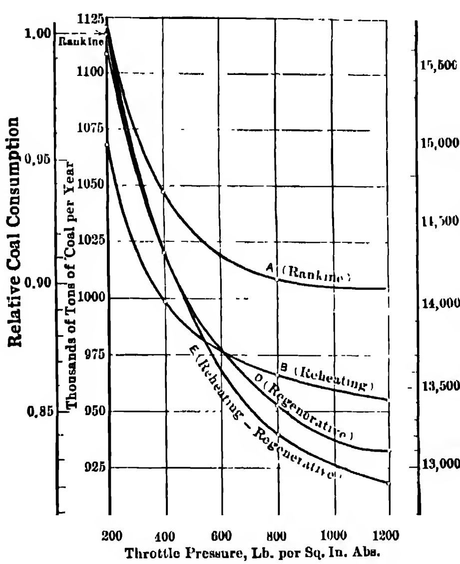


FIG. 682. Relative Coal Consumption of 200,000-kw. Plants Operating on Various Cycles and Steam Pressure.

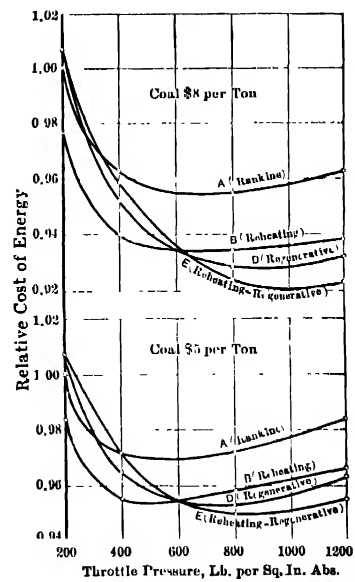


FIG. 683. Effect of Steam Pressure and Cycle on Cost of Energy at Switchboard of 200,000-kw. Plants.

exhaust pressure 1 in. Hg abs., and heating value of coal 12,300 B.t.u. per lb. Under the assumed conditions, it appears that with coal at $5.00 per ton and a pressure of 600 lb. abs., it is immaterial what cycle is used, other than the Rankine, and that the gain in economy above 600 lb. abs. is not attractive. With coal costing $8.00 per ton, the best pressure for base-load conditions would appear to be near 1000 lb. per sq. in.

The curves in Fig. 683, also by Hirshfeld and Ellenwood, show the effect of steam pressure and cycle on the probable cost of energy at the

[1] "High-Pressure, Reheating and Regenerating for Steam Power Plants.' Presented at the Annual meeting of the A.S.M.E., Dec. 3, 1923.

switchboard for this 200,000-kw. plant. Cost of energy is taken as 1 for the Rankine cycle at 200 lb. abs. pressure.

The curves in Fig. 684 by John A. Stevens and Carl J. Sittinger,[1] show the relation between coal cost, load factor, and plant costs and offer a simple means of estimating how much the necessary refinements will cost

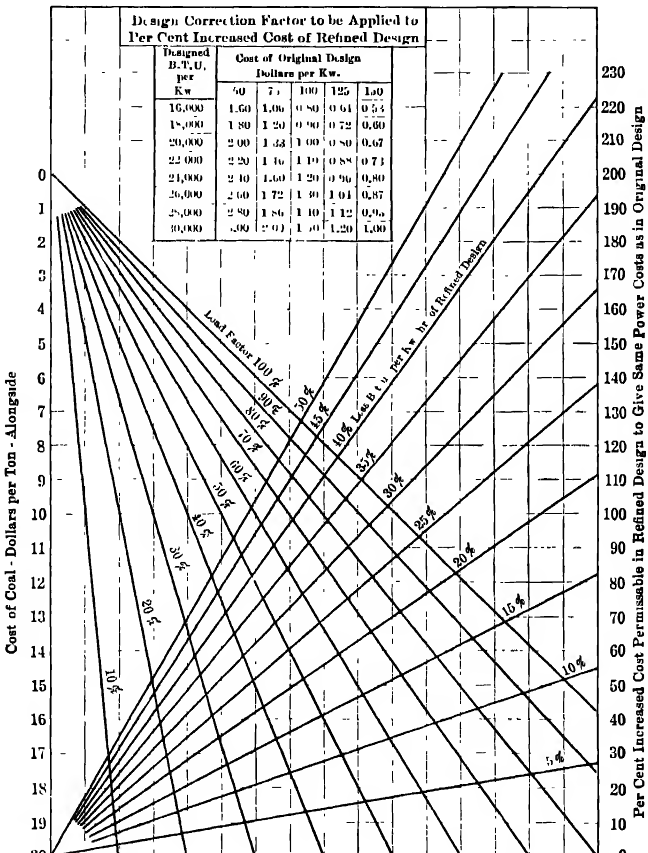


FIG. 684. Relation between Coal Cost, Load Factor and Plant Costs.

and whether the saving in operation will offset the additional fixed charges. The use of these curves is best illustrated by the examples cited by Stevens and Sittinger.

" Let us assume two initial designs, — one at $100 per kw. for a 20,000-B.t.u. station giving a correction factor of 1.00, and another at $125 per kw. for an 18,000-B.t.u. station giving a correction factor of 0.72. With

[1] Trans. A.S.M.E., Vol. 44, 1922, p. 1154.

coal at $7.50 a ton and 60 per cent load factor, how much could we afford to pay for a station giving an economy of 12,000 B.t.u. per kw-hr.?"

For the first case, the 12,000-B.t.u. refined design means 40 per cent less B.t.u. per kw-hr., for which it would be permissible to pay 62 per cent $\times$ 1.00 correction factor = 62 per cent more than the initial design, or $162 per kw.

For the second case, the 12,000-B.t.u. refined design means $33\frac{1}{3}$ per cent reduction in B.t.u. per kw-hr. for which it would be permissible to pay 51 per cent $\times$ 0.72 correction factor = 36.7 per cent more than the initial design, or $171.

Overall thermal efficiencies in the steam-electric plant vary from less than 5 per cent of the heat energy of the fuel, for the small non-condensing installation discharging the exhaust to waste, to approximately 20 per cent in the large modern condensing central station with favorable load factor. Even if it were commercially practical to install and operate a condensing plant with the most efficient cycle so far suggested, utilizing steam at 1200 lb. initial pressure, temperature 800 deg. fahr. and 1 in. abs. vacuum, the maximum possible overall efficiency would probably not exceed 30 per cent. The average small non-condensing plant in which all the exhaust steam is utilized for heating or process work is capable of furnishing electric energy at an overall heat efficiency of approximately 30 per cent, and a large non-condensing plant equipped with economical boilers and prime movers could readily realize 60 to 70 per cent efficiency under the same conditions. This naturally suggests the utilization of the exhaust of large central power stations for heating and process work, by distributing the exhaust steam directly or by circulating the condensing water through a closed system. To be of commercial value, the exhaust steam or circulating water must have a temperature much higher than that carried in the conventional condensing plant. This necessitates increased first cost of prime movers and heavy initial investment in distributing systems which, under the usual steam conditions, frequently more than offset the thermal gain. Besides, the power and heat loads do not coincide. There are a number of public utility plants operating as combined power and heating plants, in which exhaust steam is distributed at 10 to 15 lb. gage pressure, but few are giving satisfactory financial returns because of the transmission heat losses and also because of the heavy fixed charges due to the large-diameter mains required to convey low-pressure steam. With high initial pressures of, say, 1000 to 1200 lb. gage and temperatures of 750–800 deg. fahr. expanding down to 200 lb. in a non-condensing turbine or engine, it is possible to obtain a kw-hr. on a heat consumption of less than that realized in the conventional 150 lb. initial pressure, non-condensing plant exhausting at

5 to 10 lb. back pressure. The specific volume of the exhaust at 200 lb. gage will be less than one-fourth that of saturated steam at 10 lb. gage, so that a correspondingly smaller heating main may be furnished for the same capacity. This exhaust may be distributed at 200 lb. pressure and reduced to 5 or 10 lb. at the point of utilization. The non-condensing unit could float on the line and its power output could be regulated by the demand for heating steam. It could be made to operate during the heating season on an overall efficiency of 60 or 70 per cent, or more. Where conditions permit, the balance of the power could be produced by the usual condensing equipment. Whether or not such an arrangement will prove commercially successful remains to be seen, but it offers a wide field for conservation of heat. It is possible to design a steam power plant in which practically all of the calorific value of the fuel can be utilized for power and heating purposes, but the commercial success of such a design depends not only upon the cost of conserving the heat but also upon the existence of a market for its utilization.

Highest Efficiency from Industrial Steam Plants: Power, Mar. 25, 1924, p. 483.

The Margins of Possible Improvement in the Central-Station Steam Plant: Mech. Engrg., Dec., 1923, p. 685

The Benson Super-pressure Plant. Power, May 22, 1923, p. 796; May 29, 1923, p. 842.

High Pressures and High Temperatures in Central Stations: Power, July 15, 1924, p. 87.

PROBLEMS

1. The rated capacity of a turbine station is 2000 kw., annual gross output 6,380,000 kw-hr., maximum load during the year 1800 kw. Required the station load factor and the station output factor.

2. If the plant in Problem 1 costs $200 per kw. of rated capacity and the annual fixed charges amount to 14 per cent, required the fixed charges per kw-hr.

3. A plant cost originally $200,000. It is proposed to establish a sinking fund on a 3 per cent basis. If the weighted life of the plant is assumed to be 20 years and the junk value of the apparatus at the expiration of this period is estimated at 15 per cent of the original cost, how much money must be placed in the reserve fund each year?

4. What will be the accumulated fund in Problem 4 at the end of 15 years?

5. A coal-handling equipment, purchase price $25,000, is 5 years old, and the yearly operating cost, including all charges except for functional depreciation, is as follows: 1st year $2500, 2nd year $2600, 3rd year $3000, 4th year $3400, 5th year $4000. If the interest rate is 6 per cent, required the equated annual operating cost.

6. Suppose a new system is on the market, suitable for the service performed by the one discussed in Problem 5, but more economical in operation. The new system costs $35,000 and its estimated functional life and salvage are 10 years and $1500, respectively. Assuming that the salvage of the old equipment is $1200, and that the estimated equated annual operating cost of the new system is $2500, will it pay to junk the old one? Assume interest rate on the sinking-fund deposits to be 6 per cent.

7. A steam-electric power plant has been in operation for 8 years, original cost $450,000, equated annual operating cost for this period $65,000. The depreciation

annuity for this equipment is based on an assumed functional life of 20 years with interest at 5 per cent. A new and more economical plant, of the same capacity as the old one, can be purchased completely installed for \$600,000. The estimated equated annual operating expense of the new plant is \$35,000. If the old plant can be sold for \$50,000 net, what is its depreciated value? Assume a functional life of 25 years for the new unit and a salvage value of \$15,000

8. What is the depreciated value of the plant in Problem 7 on the straight-line basis? On the sinking-fund basis?

9. The average fuel consumption of a 30,000-kw turbo-generator plant is 1.8 lb. coal (11,000 B.t.u. per lb.) per kw-hr for a yearly station output-load factor of 0.42. The cost of coal is \$4 per ton of 2000 lb. and the fuel cost is 50 per cent of the total station operating costs. What is the total cost of operation, dollars per year?

10. A 20,000-kw. turbo-generator uses 11 lb. steam per kw-hr., initial pressure 215 lb. absolute, superheat 150 deg. fahr., vacuum 27.5 in. referred to a 30-in. barometer, feedwater 180 deg. fahr. If the average overall boiler and furnace efficiency is 70 per cent and the calorific value of the coal is 12,500 B.t.u. per lb., required the average B.t.u. supplied by the fuel per kw-hr. generated. Determine also the average weight of coal used per kw-hr.

11. During the winter months, all of the exhaust steam from a 500-hp. non-condensing engine-generator set is used for heating purposes. Engine uses an average of 50 lb. steam per kw-hr., initial pressure 125 lb. absolute, back pressure 17 lb. absolute, initial quality 98 per cent, feedwater 210 deg. fahr. If the average overall boiler and furnace efficiency is 65 per cent and the coal costs \$5 per ton of 2000 lb. (calorific value 12,000 B.t.u. per lb.), what is the actual cost of fuel for power only, cents per kw-hr.?

CHAPTER XX

TYPICAL SPECIFICATIONS

373. Specifications for a Horizontal Tubular Steam Boiler.[1] — The following specifications for one 72-in. horizontal-return tubular steam boiler, pressure 150 lb., were prepared by the Hartford Steam Boiler Inspection and Insurance Company for the Armour Institute of Technology, Chicago:

This specification is intended to cover the construction of one horizontal tubular boiler designed to operate at a maximum pressure of 150 lb. per sq. in. Each bidder must submit a proposal for doing the work exactly as specified, but alternate proposals involving slight modifications will also receive consideration provided such modifications are fully described.

The Boiler Contractor shall furnish the various accessories mentioned herein and he shall also provide all the necessary miscellaneous iron or steel work as hereinafter enumerated. The Contractor under this specification will not be required to construct foundations, brickwork or other masonry.

Drawings. — Drawings prepared by The Hartford Steam Boiler Inspection and Insurance Company accompany this specification and are made a part hereof; the drawings and specification are intended to supplement each other and to be mutually co-operative, and, unless otherwise noted, the Boiler Contractor shall follow all details and shall furnish all parts and fittings which may be required by the drawings and omitted by the specification, or vice versa, just as though required by both. The said drawings are identified respectively by Nos. 6260 and 4890.

General Data. — The boiler with its fittings shall be constructed and furnished in accordance with the following general data and dimensions: —

Diameter measured on inside of largest course . . . . 72 in.
Number of courses . . . . Three.
Thickness of material: Heads, $\frac{9}{16}$ in. Butt-straps, $\frac{7}{16}$ in. Shell-plates, $\frac{17}{32}$ in.
Girth seams: Single-riveted lap-joints with rivets spaced $2\frac{3}{8}$ in. on centers.
Longitudinal seams . . . . Quadruple-riveted butt-joints.
Diameter of rivets for all seams . . . . $\frac{7}{8}$ in. ($\frac{15}{16}$-in. holes).

[1] Paragraphs pertaining to properties of steel plates, rivets, and tubes have been greatly abridged because of space limitation.

Tubes: Number, 70. Diameter, 4 in. Length, 18 ft. Thickness, 0.134 in.

Braces above tubes: Number on each head, 20. Least diameter, $1\frac{1}{8}$ in. Diameter of rivet holes for attaching, $\frac{7}{8}$ in. Least cross-sectional area through sides at each rivet hole on head end, 0.55 sq. in.; ditto on shell end 1.10 sq. in.

Through-braces below tubes: Number, 2. Least diameter, 2 in. Least diameter of upset on front end, $2\frac{1}{2}$ in. Diameter of pin, $1\frac{3}{4}$ in. Least cross-sectional area through center of eye, 3.83 sq. in.

Size of blow-off pipe . . . . $2\frac{1}{2}$ in.

Diameter of nozzles: Steam opening . . . . 6 in.

Safety valve connection . . . . 6 in.

Size of feed-pipe . . . . $1\frac{1}{2}$ in.

Manholes: One in front head below tubes and one in top of shell.

Size of grates . . 72 in. long by 66 in. wide.

Height from grates to bottom of shell, at front end . 40 in.

Smoke-Box: Bolted to front head by clip angles. Smoke opening 60 in. by 14 in.

Style of Front . . . . Flush.

Fittings to be furnished with the boiler as follows: — One 10-in. steam gage graduated from 0 to 225 lb., brass siphon and union-cock for gage, two $2\frac{1}{2}$-in. safety valves with minimum lift of 0.08 in., flanged Y-base for safety valves, three $\frac{3}{4}$-in. gage cocks, one combination water-column, one $\frac{3}{4}$-in. gage glass 14 in. long.

Method of Support. — The boiler shall be suspended by means of U-bolts and steel hangers, from a framework made up of four I-beams and four columns. I-beams shall be 8 in. deep and shall weigh 18 lb. per ft.; they shall be assembled in pairs by means of tie-bolts and separators, spaced near each end and at intervals of not more than 4 ft., in such manner that the adjacent edges will be 3 in. apart. If cast-iron columns are used they shall be round with an outside diameter of 8 in. and a thickness of $\frac{7}{8}$ in., or square with a width of 8 in. and a thickness of $\frac{3}{4}$ in. Six-inch rolled-steel H-beams, weighing 23.8 lb. per ft. may be used for columns but no other form of structural steel column will be approved unless it can be shown that the safe load (figured in the usual manner with regard to length and radius of gyration) will be equal to that which can be allowed on the H-beams specified above. Steel columns shall have suitable base-plates and cap-plates riveted on and cast-iron columns shall be made with top and bottom flanges of proper design. Details of hangers, U-bolts, etc., are shown on the accompanying drawing.

Properties of Steel Plates. — (Chemical requirements have been omitted.)

Complete tests must be made to show that each plate will fulfill the above requirements in regard to tensile strength, elastic limit, chemical composition, elongation, bending, and homogeneity; and any plates failing to meet the said requirements shall be rejected. One tension, one cold-bend, and one quench-bend test shall be made from each plate as rolled. All details in regard to size and shape of specimens, method of making tests, etc., shall be in strict accordance with the "Requirements for Testing Steel," as adopted by The Hartford Steam Boiler Inspection and Insurance Company.

All tests and inspections of material may be made at the place of manufacture prior to shipment. Certified copies of reports of all tests must be approved by a representative of The Hartford Steam Boiler Inspection and Insurance Company before any of the material covered thereby is used for any portion of the work contemplated by this specification.

Stamping. — (Omitted.)

Rivets. — (Omitted.)

Details of Riveting. — Longitudinal seams shall be of the butt-joint type with double covering straps, and the details shall be as specified herein and as shown on the accompanying drawing, except that the pitch of rivets in the outer row may be increased or decreased (with corresponding changes in the pitch of rivets in the other rows) in cases where such changes are desirable in order to secure a proper spacing of rivets between girth seams. It must be understood, however, that no such change can be made without the consent and approval of the inspector having jurisdiction and no such change shall be allowed if it will result in a factor of safety lower than 5.00 or if it will produce a pitch too great for proper calking. Except for rivet holes in the ends of butt-straps, the distance from the center of the rivet to the edge of the plate must never be less than one and one-half ($1\frac{1}{2}$) times the diameter of the rivet hole. The seams must be arranged to come well above the fire-line and to break joints in the separate courses.

Rivet holes shall either be drilled full size with plates, butt-straps and heads bolted up in position or else they shall be punched at least one-quarter inch ($\frac{1}{4}''$) less than full size. If the latter method is used, plates, straps, and heads shall be assembled and bolted together after punching and the rivet holes shall be drilled or reamed in place one-sixteenth inch ($\frac{1}{16}''$) larger than the diameter of the rivets. After reaming or drilling, plates and butt-straps shall be disconnected and the burrs removed from the edges of all rivet holes. If any holes are out of true more than one-sixty-fourth inch ($\frac{1}{64}''$), they must be brought into line with a reamer or drill; evidence that a drift-pin has been used for this purpose will be sufficient cause for the rejection of the entire work. The plates must be

rolled to a true circle before drilling and the butt-straps and ends of plates forming the longitudinal joints must be formed to the proper curvature by pressure, — not by blows. Particular care must be used to secure proper fitting where the courses telescope together at girth seams. This is a matter of the utmost importance and the results obtained will be considered as a criterion of the general character of the workmanship throughout.

Rivets must be of sufficient length to completely fill the rivet holes and form heads equal in strength to the bodies of the rivets. Rivets shall be machine-driven wherever possible, and always with sufficient pressure to entirely fill the rivet holes; the authorized inspector of The Hartford Steam Boiler Inspection and Insurance Company shall have the privilege of cutting out rivets to see if satisfactory results have been obtained and all such work of cutting rivets and replacing them shall be done at the expense of the Contractor. Rivets shall be allowed to cool and shrink under pressure.

Calking and Flanging. — All calking edges shall be beveled to an angle of about fifteen degrees (15°) and every portion of such edges shall be planed or milled to a depth of not less than one-eighth inch ($\frac{1}{8}''$). Bevel-shearing will not be acceptable in place of planing or milling but chipping will be allowed in special cases provided the workmanship will meet with the inspector's approval. All seams must be carefully calked with a round-nosed tool.

Flanging must be performed in such manner that the flange will stand accurately at right angles to the face of the sheet and the straight portion of the flange must be long enough to allow for making a perfect joint with the shell plate. The radius of the bend, on the outside, shall be at least equal to four times the thickness of the head.

Tubes. — (Chemical requirements and method of testing have been omitted.) Each tube must be legibly stenciled with the name or brand of the manufacturer, the material from which it is made (steel or charcoal iron), and the words "Tested at 1000 lb."

All tests and inspections shall be made at the place of manufacture and the Boiler Contractor shall require the tube manufacturer to certify that the tubes have been tested and have met the requirements stated above. Tubes shall be rejected when inserted in the boiler if they fail to stand expanding and beading without showing cracks or flaws, or opening at the weld.

Tube holes may either be drilled full size or punched so as to have a diameter at least one-half inch ($\frac{1}{2}''$) less than full size and then drilled, reamed, or finished full size with a rotating cutter. The full size diameter of the hole shall be $\frac{1}{32}$ inch greater than the outside tube diameter. Edges of tube holes shall be properly chamfered.

Tubes shall be set with a Dudgeon expander and all ends shall be substantially beaded.

Staying. — The number, size, arrangement, and general details of stays or braces are specified on page — and shown on the drawing. No changes shall be made in the number and location of braces without the approval of The Hartford Steam Boiler Inspection and Insurance Company. All braces shall be made of solid, weldless mild steel.

Braces above the tubes shall be of the diagonal crowfoot form and none of them shall be less than three feet, six inches (3′ 6″) long. Each brace shall be attached by means of four rivets, two at each end; rivets of a larger diameter than specified on page — may be used if preferred, but the cross-sectional area through the brace at the sides of the rivet holes must be maintained as called for. Braces having a rectangular cross section may be used provided the cross-sectional area of each brace is equal to that of each of the round braces specified, and provided also that the requirements regarding size of rivets and net area through rivet holes are fulfilled. Braces must be carefully set to bear uniform tension.

Through braces shall be used below the tubes, extending from head to head. Each brace shall be upset on the rear end to form an eye and the eye shall be inserted between the outstanding legs of a pair of angle-irons and held in place by a turned bolt passing through holes drilled in both angles and in the eye. The angles shall be securely riveted to the rear head in the manner shown on the drawing, being held at a distance of three inches from the head by means of spacers made of extra heavy pipe. Spacers must be accurately squared on both ends so that they will all be of the same length and will furnish a rigid and uniform bearing for the angles. Through braces shall be upset and threaded on the front ends and shall pass through the front head, being secured with nuts and washers both inside and outside. The center line of the braces at the front head must not be lower than the center line of the manhole.

Manholes. — Manholes shall be oval or elliptical in shape, not smaller than 15 in. long by 11 in. wide, and shall conform to the following requirements: —

The manhole in the top of the shell shall be placed with its long dimension crossways of the boiler. The frame shall be made of pressed steel formed to the proper curvature, and it shall be riveted to the inside of the shell with two rows of rivets symmetrically spaced. Based on the allowance of 44,000 lb. per sq. in. the size and number of the rivets must be such that their total shearing strength will not be less than twice the tensile strength of the plate removed, as figured from the cross-sectional area in a plane passing through the center of the manhole and the axis of the shell; the net cross-sectional area of the manhole frame, as cut by

such a plane, must not be less than the cross-sectional area of the plate removed in the same plane.

The manhole in the front head shall be formed by flanging the head inwardly to a depth of not less than three times the thickness of the head all around the opening and a steel band shall be shrunk on, pinned in position, and properly machined for the gasket bearing; the band will not be required if a recessed manhole plate is used.

All necessary manhole plates, yokes, bolts, and gaskets shall be furnished to make the installation complete, the various parts being proportioned so as to have a strength equal to that of manhole frames. Manhole plates and yokes shall be made of pressed steel. Gasket bearings shall be at least one-half inch ($\frac{1}{2}''$) wide and the thickness of gaskets shall not exceed one-quarter inch ($\frac{1}{4}''$).

Nozzles. — Nozzles shall be made of pressed or cast steel and shall be of heavy and substantial design properly adapted to the pressure to be carried. They must be accurately shaped to fit the curvature of the shell and must be carefully and securely riveted in place in such manner that the face of each flange after erection will lie in a horizontal plane parallel with the upper surface of the tubes. The flange of each nozzle must be properly faced.

Feed Piping. — Feed piping must be firmly supported in the boiler in such manner that no portion of the piping can be in contact with any of the tubes or other parts of the boiler. The feed pipe shall enter the boiler through the front head by means of a brass or steel bushing placed on the left-hand side of the boiler, three inches ($3''$) above the top of the upper row of tubes as shown on the drawing. The feed pipe shall extend back from the bushing to approximately three-fifths the length of the boiler, crossing over to the center and discharging above the tubes. The pipe must not discharge in proximity to any riveted joint.

All external feed piping will be furnished under separate contract but the Boiler Contractor must leave the threads in proper condition so that the piping can be readily connected.

Blow-off Pipe Connection. — A connection for blow-off pipe shall be provided on the bottom of the shell near the rear end, as shown on the drawing. It shall consist of an extra-heavy pressed steel flange, properly tapped for the blow-off pipe and securely riveted to the boiler shell.

Fusible Plug. — A fusible plug shall be placed in the rear head, on the vertical diameter, and the center of the plug must not be less than two inches ($2''$) above the upper surface of the tubes. The plug must project through the sheet not less than one inch ($1''$).

Fusible plugs shall be filled with pure tin the least diameter of which shall be one-half inch ($\frac{1}{2}''$).

Safety Valves. — Safety valves shall be of the direct spring-loaded pop type with seats and disks of nickel or other non-ferrous material. Valves must operate without chattering and must be set and adjusted to close after blowing down not more than 6 lb. Springs must not show a permanent set exceeding $\frac{1}{32}$ in. ten minutes after being released from a cold compression test closing the spring solid; no spring shall be used for a pressure in excess of 10 per cent above or below that for which it was designed.

Each safety valve shall have a substantial lifting device with the spindle so attached that the valve disk can be lifted from its seat through a distance not less than one-tenth of the nominal diameter of the valve, when there is no pressure on the boiler.

The following items shall be plainly stamped or cast upon the body:

(*a*) The name or identifying trade-mark of the manufacturer.
(*b*) The nominal diameter with the words "Bevel Seat" or "Flat Seat."
(*c*) The steam pressure at which the valve is set to blow.
(*d*) The lift of the valve disk from its seat, measured immediately after the sudden lift due to the pop.
(*e*) The weight of steam discharged in lb. per hour at the pressure for which it is set to blow.
(*f*) The letters A. S. M. E. Std.

Safety valves having a lower lift than that specified on page — may be used but the diameter must be increased proportionately as directed by The Hartford Steam Boiler Inspection and Insurance Company.

In the absence of any specific directions from the Purchaser, the Boiler Contractor shall state in his proposal the make and style of valve which he intends to furnish. It is understood that failure to do this will give the Purchaser the right to specify the make of valve after the contract is awarded and, in such event, the Contractor agrees to furnish any make the Purchaser may select.

Fittings. — The foregoing in regard to choosing the make and style of safety-valves shall apply in the same manner and with equal force to the make of gage-cocks, water-column, steam-gage, etc.

The combination type of water-column shall be used and openings for water and steam connections must be tapped for one-and-one-quarter-inch ($1\frac{1}{4}''$) pipes. Brass pipe shall be provided for the water connection and the piping shall be made up with plugged fittings to facilitate cleaning.

The Boiler Contractor shall properly drill and tap all holes required for the installaton of the various fittings, including also a one-quarter-

inch ($\frac{1}{4}''$) pipe with valve for the connection of test gage. The sizes of steam-gage, gage-cocks, and gage-glass are specified on page —.

All nozzles, flanges, fittings, etc., furnished under this specification must correspond in diameter, drilling, and other details with the "American Standard" for the stipulated pressure.

Front. — The front shall be constructed of sectional plate steel or of cast iron and the Contractor must state in his proposal which form he intends to furnish. If made of steel, the plates must not be less than three-eighths inch ($\frac{3}{8}''$) thick (except for moldings, etc.) and they must be straight and smooth with all edges machined and properly fitted to make good joints. Heavy cast-iron door-frames with planed surfaces shall be securely bolted to the plates and the front shall be further reinforced against warping by means of channel irons or other suitable braces placed on the back.

If made of cast iron, the front must be of heavy and substantial design and all castings must be smooth, true, and free from cracks, blow-holes, or other defects.

The usual fire doors, ashpit doors, and doors for giving access to the tubes shall be provided as shown on the accompanying drawings. All doors must be of heavy design and all contact surfaces must be carefully machined so that the doors will fit closely. Each flue door must be provided with a suitable fastening at top and bottom, designed to clamp the door tightly in the closed position and prevent warping. All doors shall be furnished complete with handles, catches, hinge-bolts, etc., and fire doors shall have liner plates.

The Boiler Contractor shall furnish all necessary anchor bolts for holding the front in position and shall see that the holes for the same are properly located in the steel plates or castings. Anchor bolts shall have a diameter of at least seven-eighths inch ($\frac{7}{8}''$) and shall be threaded and provided with nuts.

All parts must be carefully made so that the front will present a neat appearance after erection. Open joints, loosely-fitting hinges or other indications of careless workmanship will be sufficient cause for rejection and the Purchaser shall have the option of making any necessary modifications and deducting the cost thereof from the contract price or of requiring the Contractor to furnish new parts which will be satisfactory.

Grates. — The Boiler Contractor shall figure on furnishing stationary grates of suitable design and shall base his proposal thereon. If requested by the Purchaser, he shall submit an alternate proposal for furnishing shaking, rocking, or dumping grates of a type which the Purchaser will specify.

Miscellaneous Iron Work. — Arch bars for rear connection shall be

made as shown on the accompanying drawings or in accordance with some detail which will meet with the approval of The Hartford Steam Boiler Inspection and Insurance Company. The Company will not approve any arch bar the metal of which is exposed to the action of the flames and hot gases.

The rear connection door must fit closely and the frame must be provided with means for anchoring into the brickwork. The door must not be smaller than sixteen inches by twenty-four inches (16″ × 24″).

The Boiler Contractor shall furnish all necessary bearer-bars for grates, all buckstays, tie-rods, lintels for clean-out doors, bolts, etc., and any other iron work, not specifically mentioned herein, which may be needed to complete the installation in the brick setting. Buckstays must be made of pressed steel or its equivalent; cast iron will not be accepted.

Tests. — The Boiler Contractor shall at all times afford all facilities to The Hartford Steam Boiler Inspection and Insurance Company, and its authorized representatives, for the test and inspection of all materials and workmanship entering into the work covered by this specification.

Hydrostatic tests shall be made in the presence of the authorized inspector of The Hartford Steam Boiler Inspection and Insurance Company and in a manner which will meet with the approval of the said inspector. The pressure for such tests shall not exceed one and one-half ($1\frac{1}{2}$) times the maximum working pressure as hereinbefore stated.

Local or State Laws. — All details of construction and installation shall be made in strict accordance with any local or State ordinances which may apply and nothing in this specification shall be interpreted as an infringement of such rules or ordinances. If any discrepancy should arise, the Contractor shall immediately report it to The Hartford Steam Boiler Inspection and Insurance Company for settlement.

374. Specifications for Steam, Exhaust, Water, and Condenser Piping for an Electric Power Station.[1] — The work referred to in this contract shall be conducted under the general supervision of ——————— (referred to as the Engineers), who shall interpret the Specifications and the Drawings that may accompany the Specifications, and shall arbitrate any controversies between the parties hereto, that may arise under this contract, their decision to be final and binding upon both of the contracting parties.

The Contractor shall comply with all laws, statutes, ordinances, acts, and regulations of the town or city, the state and the government in which the work is to be performed, and shall pay all fees for permits and inspections required thereby.

The Contractor shall, at an early date, communicate with other con-

[1] From the files of a prominent Chicago engineering firm.

tractors employed by the Purchaser, and shall work in harmony with them, any differences of opinion between contractors being arbitrated by the Engineers or their representative.

The Contractor shall begin work as soon as possible, and complete same, free of all liens and charges, on or before the time mentioned herein. If, in the opinion of the Engineers, the Contractor fails to prosecute the work with the necessary means and diligence to insure its completion within the time limit, then the Engineers shall notify the Contractor by written notice to that effect, and the Purchaser may order the Contractor to employ more men, machinery, and tools to be put upon the work, specifying the additional force required, and if the Contractor fails to comply with such written demand within six (6) days from the date thereof, or within such time as the Engineers in writing prescribe, then the Purchaser may employ necessary means to complete the work within the time required, and such additional cost caused by either the employment of additional men, machinery, or otherwise, shall be deducted from any funds due, or that may become due the Contractor on account of this contract. The Contractor shall remove any particular workman or workmen from the work, if in the judgment of the Engineers it will be for the best interest of the work.

The Engineers shall have the right to make any changes in the Drawings or Specifications that they deem desirable. Should any additional labor or material be involved in such changes, the Contractor shall be paid for supplying same; on the other hand, should such changes reduce the amount of labor or material from that originally specified, the Contractor shall sustain an equivalent reduction in the contract amount and the Engineers shall be the arbiters in determining rates of increase or reduction. No claim shall be allowed for extra labor or material above the contract amount, unless same shall have been ordered in writing, with remuneration stipulated, by the Engineers. Acceptance by the Contractor of final payment on the contract price shall constitute a waiver of all claims against the Purchaser.

All material and workmanship furnished under this contract must be of the best quality in every particular and the Contractor must remedy any defects which develop during the first year of actual service, due to faulty material or workmanship, free of expense to the Purchaser. The Purchaser, the Engineers, or their representative may inspect any machinery, material or work to be furnished under this contract and may reject any which is defective or unsuitable for the uses and purposes intended, or not in accordance with the intent of this contract, and may order the Contractor to remedy or replace same; or the Purchaser may, if necessary, remedy or replace same at the expense of the Contractor.

Until accepted in its entirety by the Purchaser, all work shall be done at the Contractor's risk, and if any loss or damage should occur to the work from fire or any other cause, the Contractor shall promptly repair or replace such loss or damage free of all expense to the Purchaser. The Contractor shall be responsible for any loss or damage to material, tools or other articles used or held for use in or about the work.

The work shall be carried on to completion without damage to any work or property of the Purchaser or of others, and without interfering with the operation of their machinery or apparatus.

The Contractor shall furnish all false work, tools and appliances that may be required to accomplish the work and shall remove all débris after erection.

The Contractor must be responsible for the safety of the work until finished and accepted by the Purchaser and must maintain all lights, guards, and temporary passages necessary for that purpose. In case of any accident causing injury to person or property, the Contractor shall obtain acquittance from or pay the injured person (whether such person be an employee, a fellow-contractor, an employee of a fellow-contractor, or otherwise) the amount of damages to which he or she may be legally entitled on account of any act or omission of the Contractor or of any agent or employee of the Contractor, during the performance of the work referred to herein, and shall provide adequate insurance to protect the Purchaser from all claims arising therefrom. The Contractor shall, further, insure the compensation provided for in any workman's compensation act which may affect the work, to all its employees or their beneficiaries, and the Contractor shall carry insurance in a company satisfactory to the Purchaser, insuring said compensation to its employees or their beneficiaries. The Contractor shall notify his insurance company and cause the name of the Purchaser to be incorporated in the compensation policy, the policy or a copy thereof to be deposited with the Purchaser upon request. The Contractor must save the Purchaser from all claims for damages set up by reason of any such injury and from all expenses resulting therefrom.

No certificates given or payments made shall be considered as conclusive evidence of the performance of this contract, either wholly or in part, nor shall any certificate of payment be construed as acceptance of defective work or improper materials. The Contractor agrees to furnish the Purchaser or the Engineers, if requested, at any time during the progress of the work, a statement showing the Contractor's total outstanding indebtedness for material and labor in connection with the work covered by this contract, such statement to be certified to by a notary public. Before final payment is made the Contractor shall satisfy the

Purchaser by affidavits or otherwise, that there are no outstanding liens for labor or materials against the Purchaser's premises by reason of any work done or materials furnished under this contract.

If, during the progress of the work, the Contractor should allow any indebtedness to accrue for labor or material to sub-contractors or others, and should fail to pay and discharge same within five (5) days after demand made by any person furnishing such labor or material, then the Purchaser may withhold any money due the Contractor until such indebtedness is paid, or apply same toward the discharge thereof.

All royalties for patents, or charges for the use or infringement thereof, that may be involved in the construction or use of any machinery or appliance referred to herein, shall be included in the contract price, and the Contractor must satisfy all demands of this nature that may be made against the Purchaser at any time.

This contract shall not be assigned nor shall any part of the work be sub-let by the Contractor without the written consent of the Engineers being first obtained, but such approval shall not relieve the Contractor from full responsibility for the work included in this contract and for the due performance of all the terms and conditions of this contract; and in no case shall such approval be granted until such Contractor has furnished the Purchaser with satisfactory evidence that the Sub-contractor is carrying ample workmen's compensation insurance to the same extent and in the same manner as is herein provided to be furnished by the Contractor.

General Data

The work herein referred to comprises the furnishing of all material and labor for the complete installation of Piping Systems for two (2) ——— kw. units to be installed in the Power Station being erected by

Each of the two (2) units is comprised of the following machinery:

(List of machinery omitted.)

All of the above machinery will be installed on the foundations by their respective contractors, and this Contractor shall make all piping connection to same unless otherwise mentioned.

Drawings. (These have been omitted.)

This contractor shall take such measurements at the building and allow for such make-up pieces as shall be necessary to make his work come true, as the Purchaser and its Engineers cannot be responsible for the exact accuracy of the dimensions given on Drawings.

The Drawings and Specifications must be taken together and any

work called for in the one or indicated in the other, or such work as can be reasonably taken as belonging to the Piping Connections and necessary to complete the system, is to be included.

Live Steam Piping

Connections from Boilers. — Each of the eight (8) boilers will be provided with two (2) 8-in. steam outlets to which this Contractor shall connect an 8-in. angle automatic stop and check valve with 7-in. outlet. From these valves Contractor shall provide 7-in. boiler leads connecting to the steam mains with gate valve at the mains, all arranged as indicated on Drawings, Nos. — and —.

Connections to Turbines. — Contractor shall provide a cast-steel manifold at rear of each of the two boilers on each unit on both sides of boiler room and connect to these manifolds the two 7-in. leads from the four boilers on each unit. From manifold at rear of boilers on north side of boiler room on each unit a 14-in. connection shall be run across the basement of firing room and connected together with 14-in. lead from manifold at rear of boilers on south side of boiler room of each unit into a 17-in. pipe, which shall be connected to the turbines. A 14-in. hydraulically operated valve shall be provided on each 14-in. line where they connect together into the 17-in. turbine lead; a gate valve shall be provided on turbine lead.

Connections shall be provided complete with cast-steel manifolds, valves, drip pockets, pipe lengths and bends, all of sizes and arranged as indicated on Drawings, Nos. -- and —.

Steam Loops. — Contractor shall provide the 12-in. steam loops between the steam leads to turbines complete with pipe bends and a hydraulically operated gate valve on each end of loop. Hydraulically operated gate valves shall also be provided for connecting the future loop, all as indicated on Drawings, Nos. — and —.

Steam to Auxiliaries. — This Contractor shall install a 4-in. auxiliary steam header along division wall between turbine and boiler rooms, with connections to manifolds at rear of boilers on south side of boiler room with gate valve at each manifold, all arranged as indicated on Drawings, Nos. — and —. From the auxiliary header connections shall be made to one service pump in condenser well, three feed pumps in boiler room, exciter in turbine room, two auxiliary oil pumps on turbines and two tempering coils on air washers, as shown on Drawings. The steam connection to each of the pumps must be provided with angle or globe throttle valve at pump. A gate valve must be provided on each connection near header, as indicated on Drawings. Each of the three (3) turbine-driven feed

pumps will be provided with a 3-in. pressure governor by Pump Contractor, which this Contractor shall install in the steam line. The steam-driven service pump shall be provided with a 2-in. pressure governor by Pump Contractor, which this Contractor shall install, providing a by-pass with three valves around same, one of which is to be the throttle valve, the other two gate valves. This Contractor shall also provide a 3-in. steam connection to the exciter, a globe valve at the turbine, and a gate valve at header.

On the steam connections to the oil pumps and air washers this Contractor must provide a 1-in. extra heavy pressure-reducing valve with by-pass around same for each unit. These shall reduce from 250 lb. to 100 lb., and a second reducing valve shall be provided on connections to air washers reducing from 100 lb. to 10 lb.

Steam from Turbines to Heaters. — The Contractor shall furnish and install the 5-in. steam connections from outlet on intermediate stage of each turbine to the auxiliary exhaust line connecting to feedwater heaters with automatic stop and check valve, regulating valve operated by thermostat in feedwater heater, set so as to heat water to about 120 deg. fahr., pressure-reducing valve and gate valve at header, as shown on Drawings. The exhaust from steam-driven auxiliaries will go to the heaters, and it is the intention to take necessary additional steam from second stage of turbine to heat the feedwater to required temperature.

Steam Connections to Soot Ejectors. — Contractor shall provide a $1\frac{1}{2}$-in. steam header lengthwise on each side of boiler room, with connections to cast-steel manifolds in main steam connections with gate valve at north side of boiler room and to auxiliary steam header with valve on south side of boiler room. From these $1\frac{1}{2}$-in. headers a 1-in. connection with globe valve having extended stem shall be run to the ejectors in basement, for each of the two divisions of each of the eight economizers, all arranged as indicated on Drawings, Nos. - , — and .

Steam Ejectors on Condenser Discharge Pipes. — Contractor shall provide a 4-in. ejector on top of each of the two (2) 54-in. condenser discharge pipes. These shall be of Schutte & Koerting or other make that Engineers may approve. To each of these ejectors Contractor shall provide a 1-in. steam connection with valves on both ends of line; also run a 4-in. discharge connection to 6-in. bilge pump discharge line with gate and check valve on each line.

Supports for Live Steam Piping. — The main supporting beams upon which the manifolds and fittings are supported will be provided by contractor for building steel, but this Contractor shall furnish the steel brackets framing to the main members above mentioned; also all roller and anchor bearings, complete with base castings, rollers, straps, springs, etc., all as

indicated and detailed on Drawings. He shall provide the steel frames for supporting the 14-in. steam load across the boiler room basement. He shall also provide the bearings for supporting the pipes on those supports. This Contractor shall also provide the main anchor bearings for the 17-in. steam loads to turbines; also the roller bearings and brackets for the 17-in. steam load to Unit No. 2.

The steel brackets for supporting the auxiliary steam header shall be provided by Contractor for Building Steel, but this Contractor shall provide the roller and anchor bearings on these brackets, all as indicated on the Drawings.

Contractor shall also provide such additional hangers, braces and supports for the steam piping as may be necessary to properly support the steam piping, and keep same free from vibration. These must in all cases be of steel or iron, and made subject to the approval of the Engineers.

Steam Drips and Drains. — The main steam headers shall be drained to the 10-in. drip pockets in boiler room basement. This Contractor shall provide and install a $1\frac{1}{2}$-in. steam trap for each unit for draining the drip pocket and must connect up same with a $1\frac{1}{2}$-in. pipe. The discharge from the trap shall be connected to the feedwater heater. Connections at trap shall be arranged with by-pass with three valves, so trap can be cut out of service.

Each of the 7-in. gate valves on steam leads from boilers shall have a boss tapped for $\frac{1}{2}$-in. drain above seat, which this Contractor shall connect into a $1\frac{1}{4}$-in. line for each unit and connect same with stop and check valve to the feedwater heater, also to the clear water reservoir; $1\frac{1}{4}$-in. lines to be cross-connected with valves. Contractor shall provide a boss tapped for $\frac{3}{4}$-in. drain on the 12-in. hydraulically operated gate valves on steam loop, also on the two 14-in. valves on lead from manifolds at rear of boilers for each unit, and connect same with a $1\frac{1}{4}$-in. pipe to their respective steam traps, providing by-pass with valves as indicated diagrammatically on drawings. The 12-in. gate valve for future steam loop shall also have boss tapped for $\frac{3}{4}$-in. drain and connected to the $1\frac{1}{4}$-in. drain line. A globe valve shall be provided on each drain connection. Contractor shall also tap the blind flange on tee in steam connection to condenser well and provide a $\frac{3}{4}$-in. drain connection with trap and discharge connection to the feedwater heater. A by-pass connection with three valves shall be provided at trap. A $\frac{1}{2}$-in. drain shall also be provided from lowest point of steam connection in condenser well to drain sump.

Contractor shall run a $\frac{3}{4}$-in. drain with valve from the steam casing of the three auxiliary turbines driving the boiler-feed pumps and the turbine

driving the exciter and connect them into a 1-in. line and run to the hot-water reservoir. Drain from casing of service pump turbine to be run to drain sump in condenser well with a valve at turbine.

Contractor shall also provide such other drip and drain connections as may be necessary to properly drain the entire system of steam connections, these to be connected as may be directed by the Engineers.

Blow-off Connections

Boiler Blow-off Connections. — Each of the eight boilers will be provided with six (6) $2\frac{1}{2}$-in. blow-off fittings on mud drums, which this Contractor shall connect up to a special fitting on each side of each boiler and from which $2\frac{1}{2}$-in. connections shall be made to the blow-off header under each row of boilers. Eight (8) $2\frac{1}{2}$-in. blow-off valves shall be provided on the blow-off connection from each of the eight boilers, all arranged as indicated on Drawings.

Contractor shall also provide the 4-in. blow-off header under each row of boilers and run 4-in. connections from same to the steel blow-off tank in boiler-room basement. This tank shall be furnished and installed by the Contractor. The Contractor shall also provide the overflow and drain connections to discharge well and vent connections to atmosphere, all of sizes and arranged as indicated on the Drawings.

Superheater Blow-off Connections. — This Contractor shall furnish and install the superheater blow-off connections from each of the eight boilers to the blow-off header in basement, as indicated on Drawings. Each boiler will be provided with two (2) 2-in. elbows and two (2) 2-in. valves, one on each end of each drum and two elbows and two valves on superheater, which this Contractor must connect to the headers. Six (6) 2-in. valves must be provided for these connections on each boiler, all arranged as indicated on Drawings.

Blow-off from Economizers. — Each of the eight (8) economizers will be provided with eight (8) $2\frac{1}{2}$-in. blow-off outlets, provided with angle valves. This Contractor shall connect these together to a 4-in. header, providing a $2\frac{1}{2}$-in. valve on each of the two divisions on each of the eight economizers. Headers shall be run along just below economizer floor, and 4-in. connection shall be run to hot-water reservoir and 4-in. to discharge line from blow-off tank. A globe valve with extended stem shall be provided on each of these connections. A check valve shall also be provided where connection is made to discharge from blow-off tank. The tee on the economizer side of these globe valves shall be tapped for $\frac{1}{2}$-in. pipe and the connection run to a pet cock above boiler-room floor, which shall drain into a funnel connected to discharge well.

Exhaust Connections

Exhaust Connections from Turbines. — This Contractor shall furnish and install the 42-in. free air exhaust connections from each of the two (2) turbines, as indicated on Drawing No. —, made up of cast-iron pipe and fittings and riveted steel pipe with forged steel riveted flanges, as made by the American Spiral Pipe Works. The steel pipe shall be close riveted and thoroughly calked so as to be air and water tight. Copper expansion joint shall be provided between main turbine exhaust and relief valve on each unit. The vertical risers shall be of $\frac{1}{8}$-in. plate and shall terminate above roof, with hoods over same, as per detail on Drawings. Horizontal pipe between relief valve and base elbow shall be of $\frac{3}{16}$-in. steel plate. There is to be no longitudinal seam on bottom of this pipe. The exhaust relief valves in these lines shall be as hereinafter specified under "Material and Workmanship."

Exhaust Connections from Auxiliaries. — This Contractor shall connect up the exhaust outlet on the three (3) turbine-driven feed pumps, auxiliary oil pumps, service pump and exciter together, and make connection to each of the two feedwater heaters, with gate valve at each pump, each heater and sectionalizing valve between heaters, all of sizes and arranged as indicated on Drawings. A 10-in. riser to atmosphere with combination back pressure and relief valve near heater and —— exhaust head above roof shall be provided on connections to each of the two heaters. Exhaust heads shall be of No. 16 galvanized iron and of most improved type. Each heater will also be provided with a 4-in. relief outlet, which this Contractor shall connect up with a back pressure valve to the 10-in. relief pipe to atmosphere on each unit, all arranged as indicated on Drawings.

Heating System for Switch House, Operating Room and Offices. — Contractor shall furnish and install for heating switch house, operating room, and offices, a complete two-pipe heating system, with overhead supply system and drain in basement. The switch house heating system shall have a total direct radiation of approximately 1,912 sq. ft., divided into 17 radiators. The operating room, offices, bedrooms, stair hall, etc., at end of turbine room shall have a total radiation of approximately 3188 sq. ft., divided into 55 radiators, all of sizes and arranged as may be directed by the Engineers. A layout drawing showing size of radiators and sizes of branch connections will be provided later. All radiators to be "————————————" two-column radiators, or other make that the Engineers may approve. All radiators to have top steam connections.

Steam for this system shall be taken from the auxiliary exhaust header

in boiler room, with a 6-in. connection running up the stair hall to the bus chamber under switch house, with gate valve and 3-in. safety valve set at 5 lb. pressure in boiler room. A low-pressure header shall be run across the bus chamber and up to the overhead header in switch house, which shall be run along the south wall and connected to the radiators in switch house. An overhead line shall also be run around three sides of the office space over switchboard room with drop connections to the radiators on the different floors. Drains from the radiators shall all be brought together and connected to a direct-connected, geared, motor-driven vacuum pump as made by the American Steam Pump Co. and of ample capacity for the service and to maintain a vacuum of 5 in. at the outlet of radiators. Motor to be similar to those hereafter specified and must be complete with starting equipment switches, fuses, etc. All wiring between motor and equipment to be provided.

Discharge from pump shall be connected to the feedwater heater by means of a float-controlled vent, as made by ———— Company.

A $\frac{1}{2}$-in. siphon trap shall be provided on outlet of each radiator, as made by ————, and a standard radiator valve provided on inlet of each radiator. All piping to be rigidly suspended in approved manner.

Safety Valve Vent Pipe. — This Contractor shall furnish and install the safety valve vent pipes on each of the eight (8) boilers, as shown on Drawings, Nos. ———. The Discharge openings of the six (6) $4\frac{1}{2}$-in. safety valves on drum of each boiler shall be connected together as indicated, and a 12-in. riser run through roof and terminating in a 12-in. tee. He shall also furnish and install the safety valve vent pipes from the discharge openings on each of the two (2) 4-in. superheater safety valves on each of the eight (8) boilers. The outlets of two valves shall be combined into a 6-in. pipe and run through roof terminating in a 6-in. tee. A $\frac{1}{2}$-in. drain pipe shall be provided on elbows at each safety valve, connecting into a $\frac{3}{4}$-in. pipe from each boiler, which shall be run to ashpit.

Exhaust Drips. — This Contractor shall install a $2\frac{1}{2}$-in. drip pipe from the 42-in. free exhaust from each turbine, providing a deep U-trap and discharging into hot-water reservoir under boiler room basement floor.

The Turbine Contractor will connect up the drains from the carbon packing rings into a 3-in. pipe on each of the two (2) turbines. This Contractor shall connect each of these pipes to the hot-water reservoir. Gate valves on vertical connections from auxiliaries shall be tapped above seats for $\frac{1}{2}$-in. bleeders, which shall be connected together into a 1-in. line and run to hot-water reservoir. Drain from gate valve on service pump shall be run to drain sump in condenser well.

Support for Exhaust Piping. — Relief valves on turbine exhaust lines

shall be provided with bases, which will be supported from floor under valves, and the vertical risers will be carried on the base elbows, but this Contractor shall provide and set angle iron braces for vertical risers, as per detail.

This Contractor shall provide all necessary anchors, hangers, and braces for properly supporting the auxiliary exhaust lines, as may be required by the Engineers.

Water Piping

Circulating Water Connections. — Purchaser shall provide and install the suction connection from intake crib to the suction inlet on each of the two circulating pumps.

Condenser Contractor shall provide the discharge connection from circulating pump to condenser on each unit.

Purchaser shall furnish and install the condenser discharge piping outside of condenser well, including gate valves, elbows, and vertical pipe length in discharge well, but this Contractor shall provide the special fitting, pipe lengths, and expansion joints on condenser discharge connections inside of condenser well. One of the pipe lengths on discharge connection from Unit No. 1 in the condenser well will be provided on ground by Purchaser, but this Contractor shall install same, providing gaskets and bolts for making up joints, all arranged and of sizes as indicated on Drawing ——. Contractor shall also provide the 6-in. tail pipes from 54-in. gate valves in discharge well.

Hotwell Pump Connections. — Contractor shall connect up the two hotwell pump discharge outlets on each unit to the inlet on primary heater in upper section of condenser, providing check and gate valve at each pump. From outlet of primary heater, connection shall be run to inlet on top of heater of each unit. The primary heater is also to be by-passed with necessary valves, all of sizes and arranged as indicated on drawings, Nos. ————. Connections to heaters shall be cross connected with valves as indicated on Drawings.

Feed-Pump Suction Connections. — Contractor shall furnish and install the suction connections to the two (2) feed pumps on each unit with connections from heater, filtered water header and unfiltered water system with valve on each connection, all of sizes and arranged as indicated on Drawings, Nos. ————. Suction connections from heaters shall be cross connected with valve as indicated.

Boiler-Feed Piping. — This Contractor shall furnish and install discharge connections from the feed pumps to the feed headers and from feed headers to economizers and boilers, all arranged as shown on Draw-

ings. There are to be two separate feedwater systems for each unit with independent connections from pumps to boilers, as shown. The auxiliary feed header is to be run in the boiler room at rear end between boilers and in basement across firing room to boiler on north side of room, with connections from same to boilers. The main feeder header shall be suspended from the economizer floor framing over boilers with connections to each of the eight (8) economizers and from economizers to the boilers. Connections between the economizer divisions will be provided by Economizer Contractor.

Each boiler will have two (2) feed inlet connections and Boiler Contractor will provide a 4-in. automatic stop and check valve on each of these outlets, to which this Contractor shall connect.

Each economizer will be provided with a 4-in. inlet at bottom and a 4-in. outlet at top, which this Contractor shall connect up.

From the 7-in. auxiliary feed headers, this Contractor shall run a 4-in. connection up the front of boilers, with a 4-in. connection to the inlet at each end of drum, providing a gate valve at header connection and a globe and check valve in horizontal run at front of boiler.

From 7-in. main feed headers, Contractor shall make a 4-in. connection to each economizer with two gate valves on each connection. He shall also make a 4-in. connection from outlet of each economizer to the feed line connecting to each of the boilers, providing a gate and check valve at economizer outlet and an angle globe valve with extended stem all arranged as indicated on Drawings.

Contractor shall provide two air chambers on each of the two main feed headers, and one air chamber on each of the two auxiliary headers, with gate valve on headers and with compressed air connections with extra-heavy stop and check valves.

Contractor shall provide a 6-in. cross connection between the two (2) 7-in. main feed lines and auxiliary feed lines, with gate valve on each connection, as indicated. Connections at pumps shall be arranged with special two-way check valves and gate valve, all of sizes and arranged as indicated on Drawings.

Water Connections to Hydraulically Operated Valves. — This Contractor shall provide and connect up a four-way cock for the hydraulically operated valve on the steam lead to turbine; the two 14-in. valves on steam lead from boilers; the 12-in. valve on steam loop on each unit and the 12-in. valve for future steam loop. The 4 four-way cocks on each unit are to be located in a box set in the division wall between boiler and turbine rooms, all as indicated on Drawings. Boxes shall also be provided by this Contractor. Water supply for the four-way cocks is to be taken from both the feed headers, with gate and check valves arranged as in-

dicated on Drawing. Drain connections with troughs and drain pipes connected to hot-water well are to be provided as indicated.

The following items included in the complete specifications have been omitted:

High-pressure Boiler-Washing System.
Service Water Piping.
Make-up Water Connections.
Water Drains.
Miscellaneous Drains and Vents.
Oil Connection to Turbines.
Pipe and Fittings for Oiling Systems.
Compressed-air System.
Air Washer Circulating Pump Suction.
Floor and Wall Thimbles.
Hose.
Thermometers and Gages.

Material and Workmanship

General Instructions. — All material and workmanship supplied under these Specifications shall be the best of their respective kinds.

All material shall be such as specified herein and free from defects or flaws of any kind, and subject to such tests and requirements as may be herein described or as may be necessary to prove the effectiveness of the material or workmanship. All labor is to be performed by men skilled in their particular line of work, and to the full satisfaction of the Supervising Engineers or their representatives. The Specifications contemplate the very best quality of material and the most mechanical character of workmanship.

All of the work shall be erected, ready for practical use, to the satisfaction of the Engineers, and all bolts, gaskets, and necessary adjuncts shall be furnished by this Contractor.

This Contractor shall satisfy himself as to the accuracy of the Drawings, and must take such measurements and allow for such make-up lengths or pieces as may be necessary to make his work come accurately together. The piping must be erected so as to preserve accurate alignment and no iron gaskets or fillers will be allowed between flanges.

Where the work of this Contractor connects to that of another, the connections shall be made by this Contractor, and he must see that all flanges for connection to the other work are properly drilled to fit the latter, irrespective of drilling dimensions on the Drawings or herein given.

The work contemplated herein shall be carried on so as to harmonize and not interfere with the work of other contractors or with the operation of the Station or any of the machinery that may be contained therein. Where connections are made to the old work, they shall be done at such time as shall meet the approval of the Chief Engineer of the Station. The work shall be installed as expeditiously as possible and subject to the general direction of the authorized Engineers.

The following items pertaining to material and construction details are included in the complete specifications but have been omitted from this copy.

- Steel Pipe.
- Welded Flanges.
- Threaded Flanges and Unions.
- Fittings.
- Valves.
- Hydraulically Operated Valves.
- Relief Valves.
- Special Valves and Appliances.
- Traps.
- Flanged Joints.
- Cast-iron Pipe.
- Supports and Hangers.
- Testing.
- Pipe Covering.
- Painting.

Condenser Specifications and Bids for Detroit Municipal Plant: Power, Dec. 11, 1923, p. 934.

Fuel Specifications: Report of Prime Movers Committee, N.E.L.A., June, 1924.

Specifications. Report of Prime Movers Committee, N.E.L.A., Feb., 1926.

CHAPTER XXI — Supplementary

PROPERTIES OF SATURATED AND SUPERHEATED STEAM

375. General. — The thermal and physical properties of water vapor, though based on experimental data, permit of accurate mathematical formulation, but the equations involved are too complex and unwieldy for everyday use. Tables and graphical charts calculated and plotted from these laws offer a simple and accurate means of solving practically all steam problems, and recourse to thermodynamic analysis is seldom necessary.

Several tables and graphical charts of the properties of saturated and superheated steam have been published, and though the values given by the various authorities differ somewhat from each other the variation is negligible for most engineering purposes, at least for pressures under 250 lb. abs. The steam tables of Peabody,[1] Marks and Davis,[2] and of Goodenough[3] are most commonly used in American engineering practice. These tables give the simultaneous physical and thermal properties of saturated and superheated steam for various pressures and temperatures. All three tables are practically identical in arrangement as far as saturated steam is concerned but differ somewhat in the treatment of superheated steam. At this writing (1926) the Bureau of Standards, Mass. Inst. of Technology, and Harvard University are engaged in research work with a view of perfecting and extending steam tables, a progress report of which may be found in Mech. Engrg., Jan., 1926, p. 144.

376. Notations. — It is to be regretted that there is no accepted standard set of symbols for designating the various properties of steam. The use of different notations for the same property, as in the tables under consideration, leads to much confusion. In the following discussion an attempt has been made to follow general practice rather than that of any particular author.

377. Standard Units. — The mean B.t.u., or $\frac{1}{180}$ of the heat required to raise one pound of water from 32 deg. to 212 deg. fahr., is the accepted standard heat unit in all recent works on thermodynamics.

[1] Steam and Entropy Tables, Peabody, John Wiley & Sons, 1909.

[2] Steam Tables and Diagrams, Marks & Davis, Longmans, Green & Co., 1912.

[3] Properties of Steam and Ammonia, Goodenough, John Wiley & Sons, 1915.

The mechanical equivalent of heat J may be taken for all engineering purposes as

1 mean B.t.u. = 778 standard ft.-lb.
(Goodenough, J = 777.64; Marks & Davis, J = 777.54.)

The reciprocal of J or $\frac{1}{778}$ is generally designated by the letter A.

The value of the absolute zero has been variously given as ranging from 459.2 to 460.66 deg. fahr. below zero. The most generally accepted value is 459.6. For all engineering purposes, the value 460 degrees is sufficiently accurate. Temperatures referred to zero deg. fahr. are generally designated by t and absolute temperature by T.

The normal pressure of the atmosphere, or one standard atmosphere, is taken as 29.921 in. of mercury at 32 deg. fahr., or 14.6963 lb. per sq. in. For most purposes these values may be taken as 30 in. of mercury at ordinary room temperature and 14.7 lb. per sq. in., respectively. Steam pressure should always be stated in absolute terms and not "gage" since the atmospheric pressure varies within wide limits. Notations p and P are commonly used to designate pressure, but because of the various methods of measuring this property they should be qualified to this effect. In the following discussion p represents pounds per square inch absolute and P pounds per square foot absolute.

378. Quality. — This term applies strictly to the per cent of vapor in a mixture of vapor and water or *wet* steam, and is usually designated by x; thus a quality of 0.95 signifies that 95 per cent of the total weight of the mixture is vapor. For saturated steam x = 1. The quality of superheated steam is designated by the temperature of the vapor or the degrees of superheat. The latter term refers to the difference between the actual temperature and that of saturated vapor of the same pressure.

379. Temperature-Pressure Relation. *Saturated Steam.* — All properties of saturated steam depend on temperature only. For any temperature there is a corresponding pressure, the relationship being determined from formulas based upon experimental data. A large number of formulas have been proposed to represent this relationship, but the more exact equations are too cumbersome for everyday use. In Marks and Davis' steam tables the pressure-temperature relationship is based upon the following law:

$$\log p = 10.51535 - 4873.71\ T^{-1} - 0.00405096\ T + 0.00000139296\ T^2. \qquad (306)$$

Wet Steam. — The relation between pressure and temperature is the same for wet steam as for saturated, since the quality does not affect the temperature.

Superheated Steam. — The temperature of superheated steam is not de-

pendent solely upon the pressure and some additional property is necessary to fix the relationship.

380. Specific Volume. *Saturated Steam.* — The specific volume s of saturated steam, or the number of cubic feet occupied by one pound, varies with the pressure and is equal to the sum of the original volume of one pound of water σ, and u the increase in volume during vaporization, thus:

$$s = u + \sigma. \tag{307}$$

Goodenough's modification of Linde's equation is

$$u = 0.59465 \frac{T}{p} - (1 + 0.0513\, p^{\frac{1}{4}}) \frac{m}{T^4}, \tag{308}$$

$$\log m = 10.825.$$

Wet Steam. — The specific volume v of wet steam may be calculated as follows:

$$v = xs + (1 - x)\,\sigma \tag{309}$$

$$= xu + \sigma. \tag{310}$$

s is given in all saturated steam tables. σ varies from 0.0161 cu. ft. per lb. at a pressure of 1 lb. per sq. in., absolute, to 0.02 cu. ft. at 300 lb. σ is so small compared with s that it may be neglected for most purposes and the specific volume becomes $v = xs$. v may be taken directly from the volume-entropy chart.

Superheated Steam. — The specific volume of superheated steam v' is given in all superheated tables. The values in Goodenough's tables were calculated from equation (308) by substituting $u = v' - \sigma$.

Wm. J. Goudie (*Engineering*, July 1, 1901) gives the following simple rule for determining the specific volume which gives satisfactory results for moderate degrees of superheat.

$$= s\,(1 + 0.0016\, t'), \tag{311}$$

in which

s = specific volume of saturated steam, cu. ft. per lb.,
t' = degree of superheat.

Tumlirz's formula is a simple and fairly accurate abridgment of equation (308) for moderate degrees of superheat but at higher temperatures gives results that are too low.

$$v' = 0.5962 \frac{T_{\text{sup.}}}{p} - 0.256. \tag{312}$$

381. Heat of the Liquid. — The heat of the liquid q, B.t.u. per lb. above 32 deg fahr., is the amount added to water at 32 deg. fahr. in order to bring it to the temperature of vaporization, thus:

$$q = \int_{492}^{T} c\, dT, \tag{313}$$

in which c = specific heat at constant pressure.

c varies with the temperature, but the relationship does not permit of simple formulation. If c_m = mean specific heat for the temperature range,

$$q = c_m (t - 32). \tag{314}$$

For many purposes it is sufficiently accurate to assume $c_m = 1$, then $q = t - 32$. The relationship between t, c, and c_m is shown in Fig. 685 for a wide range in temperatures.

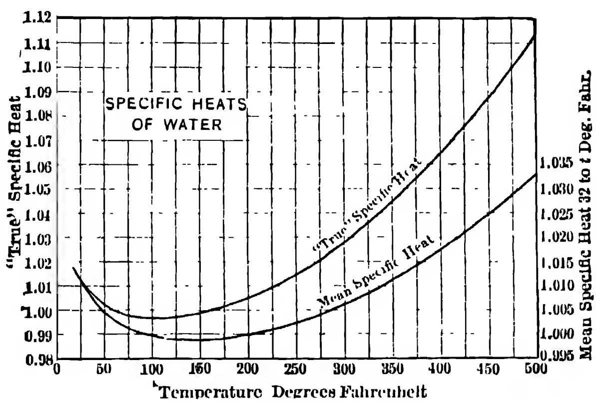


FIG. 685. Specific Heats of Water.

The heat of the liquid is manifestly constant for a given temperature whatever may be the condition of the steam.

382. Latent Heat of Vaporization. — The latent heat of vaporization r, B.t.u. per lb., is the amount of heat required to change the fluid from a liquid to vapor at the same temperature. The latent heat has been accurately determined by direct experiment from 32 deg. to 356 deg. fahr. and numerous formulas have been based upon the experiments for calculating this quantity. Goodenough's values are calculated from the Clapeyron relation:

$$r = A (s - \sigma)\, T \frac{dp}{dT}. \tag{315}$$

A simple formula which gives accurate results from 32 deg. to 400 deg. fahr. is

$$r = 970.4 - 0.655\,(t - 212) - 0.00045\,(t - 212)^2. \tag{316}$$

At higher temperatures Hennings' exponential formula as modified by Dr. Davis is perhaps more accurate than equation (316),

$$r = 139\,(689 - t)^{0.315}. \tag{317}$$

The latent heat decreases with the increase in temperature until a temperature of approximately 706 deg. fahr. (corresponding pressure 3200 lb. per sq. in.) is reached, when its value becomes 0. This is called the *critical* temperature.

Values of r are given in all saturated steam tables.

Special interest attaches to the values of r at 212 deg. fahr. because of its common use in engine and boiler tests. The following values have been assigned to this quantity.

Regnault	966.0	Marks and Davis	970.4
Peabody	969.7	Smith	972.0
Heck	971.2	Goodenough	971.6

External Latent Heat. — During the heating of the liquid the change in volume is very small and may be neglected; hence the external work done is negligible and also practically all of the heat goes to increase the energy of the liquid. During vaporization, however, the volume changes from σ to s. Since the pressure remains constant, the external work that must be done to provide for increase in volume is

$$P\,(s - \sigma) = Pu \tag{318}$$

and the corresponding heat or external latent heat is

$$AP\,(s - \sigma) = APu. \tag{319}$$

Internal Latent Heat. — The heat r added during vaporization is used in increasing the energy and is doing external work. Hence the difference, or internal latent heat ρ, B.t.u. per lb. above 32 deg. fahr.,

$$\rho = r - APu, \tag{320}$$

is the heat required to do disgregation work.

383. Total Heat or Heat Content.[1] — *The total heat of saturated steam* λ, B.t.u. per lb. above 32 deg. fahr., is evidently the sum of the heat of

[1] The heat content or *initial thermal potential* is greater than the total heat by the heat equivalent of the work of pumping the liquid into the boiler, $AP\sigma$. This quantity is negligible for most practical purposes. Modern steam tables give heat content rather than total heat.

the liquid and the heat of vaporization, or

$$\lambda = r + q \tag{321}$$
$$= \rho + APu + q. \tag{322}$$

The *heat content* of saturated steam may be calculated by means of the Davis formula:

$$\lambda = 1046.187 + 0.6077\, t - 0.00055\, t^2. \tag{323}$$

The quantity $(\rho + q)\frac{1}{A}$ gives the increase in energy of the saturated vapor over that of the liquid at 32 deg. fahr. and is called the *intrinsic energy*.

Wet Steam. — If vaporization is not complete the heat content H_w, B.t.u. per lb. above 32 deg. fahr., may be expressed:

$$H_w = xr + q \tag{324}$$
$$= x\rho + APxu + q. \tag{325}$$

Superheated Steam. — If heat is added at constant pressure after vaporization is completed, the vapor will be superheated, and the heat content H_s is

$$H_s = r + q + C_m t' \tag{326}$$
$$= \lambda + C_m t', \tag{327}$$

in which

C_m = mean specific heat of the superheated vapor at constant pressure,
t' = degree of superheat = $t_{\text{sup}} - t_{\text{sat}}$.

Goodenough gives the following formula for calculating the heat content of superheated steam, absolute temperature of the steam T_s deg. fahr.

$$H_s = 0.320\, T_s + 0.000063\, T_s^2 - \frac{23{,}583}{T_s} - \frac{C_3 p\,(1 + 0.0342\, p^{\frac{1}{2}})}{T_s^4}$$
$$+ 0.00333\, p + 948.7, \tag{328}$$

$\log C_3 = 10.7915.$

384. Specific Heat of Steam. *Saturated Steam.* — If the amount of heat required to raise the temperature of saturated steam one degree and still maintain a saturated condition is construed as the specific heat of saturated steam, then the quantity is negative, since heat must be abstracted to effect this result.

$$C_{\text{sat.}} = 0.35 - 0.000666\,(t - 212) - \frac{r}{T}. \tag{329}$$

Superheated Steam. — The *true* or *instantaneous* specific heat C' of superheated steam at constant pressure is the amount required to increase the temperature of one pound one degree fahr. Goodenough's equation based on the experiment of Knoblauch and Jakob is

$$C' = 0.320 + 0.000126\ T_s + \frac{23{,}583}{T_s^2} + \frac{C_2 p\ (1 + 0.0342\ p^{\frac{1}{2}})}{T_s^5}, \qquad (330)$$

$\log C_2 = 11.3936$.

The *mean* specific heat may be calculated from superheated steam tables as follows:

$$C_m = \frac{H_{\text{sup}} - \lambda}{t'}. \qquad (331)$$

The mean specific heat of superheated steam at constant pressure for a wide range in pressures and temperatures is given in Table 135. The values are calculated from Marks and Davis' Steam Tables.

385. Entropy. *General.* — No change in a system of bodies that takes place of itself can increase the available energy of the system. As a matter of fact, the actual physical process is accompanied by frictional effects and the quantity of energy available for transformation into work is decreased. This decrease in available energy or increase in unavailable energy is given the name *increase of entropy.* Although the solution of all engineering problems involving thermodynamic changes can be obtained without employing entropy, still its use simplifies the calculation in much the same manner that logarithms facilitate complex numerical computations. Increase of entropy between the absolute temperatures T_2 and T_1 may be expressed mathematically

$$\text{Increase of entropy} = \int_{T_2}^{T_1} \frac{dQ}{T}, \qquad (332)$$

in which dQ represents an infinitesimal amount of heat and T the absolute temperature at which it is added.

Entropy of the Liquid. — The increase in entropy θ due to heating one pound of liquid from 32 deg. fahr. to temperature T is

$$\theta = \int_{492}^{T_1} \frac{dq}{T} = \int_{492}^{T_1} \frac{c\,dT}{T}, \qquad (333)$$

in which

T_1 = absolute temperature of the liquid = $t_1 + 460$,

q = heat of the liquid above 32 deg. fahr., B.t.u. per lb.,

c = specific heat of water at temperature T.

TABLE 135

MEAN SPECIFIC HEAT OF SUPERHEATED STEAM

(Computed from Marks and Davis' Steam Tables)

Absolute Pressure, Pounds per Square Inch	Degrees of Superheat, Fahr.																			
	10	20	30	40	50	60	70	80	90	100	110	120	130	140	150	175	200	225	250	300
1	452	.452	453	453	454	454	454	455	455	455	455	455	455	456	456	.456	456	456	456	.457
5	460	460	460	460	.460	460	459	459	459	459	459	459	459	459	459	459	460	460	460	460
10	465	465	.465	465	464	.464	464	464	464	464	464	464	464	464	464	464	465	465	465	465
15	470	470	470	470	470	470	469	469	469	469	469	469	469	469	469	468	468	468	468	.468
20	475	475	475	475	474	474	.474	474	.473	473	473	473	473	472	472	472	472	472	471	.471
25	480	480	480	479	479	479	478	478	478	477	477	477	477	477	.476	476	476	475	475	.475
30	485	485	484	484	484	483	483	.483	483	482	482	481	481	481	480	480	479	479	478	477
35	490	.490	489	489	.488	488	487	487	486	486	486	485	485	484	484	483	.482	482	481	480
40	495	494	494	493	492	492	491	491	.490	490	489	489	488	488	487	486	485	484	483	482
50	509	508	507	506	504	503	501	500	499	498	497	496	496	495	494	493	491	490	489	487
75	540	535	533	530	528	525	523	521	519	517	515	513	512	511	509	506	504	502	500	497
100	570	560	556	552	550	546	542	540	537	534	531	528	526	524	522	518	515	512	509	505
125	590	.585	580	575	570	565	560	556	552	548	545	542	539	537	534	532	524	520	517	510
150	620	615	606	597	592	585	579	572	566	562	558	554	550	547	544	539	533	528	524	518
175	660	645	633	622	614	605	597	589	582	577	572	566	562	558	555	548	541	535	531	524
200	.690	675	.657	.645	634	623	614	605	596	590	584	578	572	568	.563	556	548	542	537	529
225	730	710	.686	672	656	643	631	620	611	603	595	589	583	578	573	.562	553	547	543	535
250	770	740	712	695	678	663	648	635	624	615	606	599	593	587	.582	570	562	555	549	540
275	800	770	.746	722	700	681	664	650	638	629	620	610	602	596	590	578	569	561	555	545
300	.850	805	773	740	724	702	683	666	.652	641	630	621	.612	.605	599	586	576	568	561	550
350	.950	890	843	802	770	741	717	697	680	666	654	.642	632	624	617	602	590	.580	573	560
400	1 06	975	913	860	818	783	754	730	709	692	677	665	654	.644	635	618	604	.594	585	571
450	1.20	1 10	1 00	925	860	816	771	762	744	720	700	683	677	.664	653	632	620	607	596	580
500	1 30	1 15	1 07	975	920	866	830	800	766	750	.727	708	692	686	673	654	635	620	608	593
550	1 50	1.30	1.17	1.07	1 00	933	885	.837	811	780	754	.741	722	.707	.700	.670	.655	.636	624	607

Since c varies with the temperature according to a rather complex law, the integration in equation (333) does not reduce to a simple form. For example, Goodenough's equation for the range 32–212 deg. fahr assumes the form

$$\theta = 2.3623 \log T + 0.0045775 \log (t + 4) - 0.00022609\ T \\ + 0.00000012867\ T^2 - 6.28787. \tag{334}$$

If the value of the mean specific c_m is known for the given temperature range, equation (333) reduces to the simple form

$$\theta = C_m \log_e \frac{T}{492}. \tag{335}$$

Values of θ are found in all unabridged steam tables.

Entropy of Vaporization. — Since the temperature at which vaporization takes place is constant, the change of entropy experienced by the fluid during vaporization is

$$n = \frac{Q}{T} = \frac{r}{T}. \tag{336}$$

If vaporization is incomplete as in case of wet steam,

$$n_w = xn = \frac{xr}{T}. \tag{337}$$

Entropy of Superheat. — The entropy change during superheating may be expressed

$$n_s = \int_{T_v}^{T_s} \frac{C'' \, dT}{T}, \tag{338}$$

T_v = temperature of the vapor.

If the value of the mean specific heat C_m for the temperature range T_v to T_s is known, the integration of equation (338) reduces to the simple form

$$n_s = C_m \log_e \frac{T_s}{T_v}. \tag{339}$$

Total Entropy of Saturated Steam. — The increase in entropy from liquid at 32 deg. fahr. to saturated vapor at temperature T is

$$N = n + \theta = \frac{r}{T} + \theta. \tag{340}$$

Total Entropy of Wet Steam.

$$N_w = xn + \theta = \frac{xr}{T} + \theta. \tag{341}$$

TABLE 136.

PROPERTIES OF SATURATED STEAM.* (Marks and Davis.)

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work	Heat Equivalent of External Work	Entropy of the Liquid.	Entropy of the Vapor.	Total Entropy	Specific Volume.	Density Weight per Cubic Foot, Pounds.
P	t	q	r	$\lambda = r - q$	ρ	Apu	θ	$\frac{r}{T}$	$\theta + \frac{r}{T}$	S	γ
† 0.1	35.03	3.05	1071.7	1074.7	1017.3	54.4	0.0062	2.1666	2.1728	2935.0	0.000340
† 0.2	53.15	21.23	1061.6	1082.8	1005.2	56.5	0.0423	2.0704	2.1127	1524.0	0.000656
† 0.3	64.49	32.57	1055.3	1087.9	997.7	57.6	0.0640	2.0135	2.0775	1041.0	0.000961
0.4	72.91	40.95	1050.6	1091.6	992.4	58.5	0.0800	1.9730	2.0530	794.0	0.001259
0.5	79.68	47.71	1047.0	1094.6	987.6	59.3	0.0926	1.9413	2.0332	642.0	0.001555
0.6	85.32	53.34	1043.8	1097.1	983.9	59.9	0.1029	1.9155	2.0184	541.0	0.001850
0.7	90.18	58.18	1041.1	1099.3	980.7	60.4	0.1117	1.8936	2.0053	467.0	0.002143
0.8	94.46	62.45	1038.7	1101.2	977.8	61.0	0.1195	1.8747	1.9942	412.0	0.002431
0.9	98.33	66.31	1036.6	1102.9	975.2	61.4	0.1265	1.8578	1.9843	367.9	0.002719
1	101.83	69.8	1034.6	1104.4	972.9	61.7	0.1327	1.8427	1.9754	333.0	0.00300
2	126.15	94.0	1021.0	1115.0	956.7	64.3	0.1749	1.7431	1.9180	173.5	0.00576
3	141.52	109.4	1012.3	1121.6	946.4	65.8	0.2008	1.6840	1.8848	118.5	0.00845
4	153.01	120.9	1005.7	1126.5	938.6	67.0	0.2198	1.6416	1.8614	90.5	0.01107
5	162.28	130.1	1000.3	1130.5	932.4	68.0	0.2348	1.6084	1.8432	73.33	0.01364
6	170.06	137.9	995.8	1133.7	927.0	68.8	0.2571	1.5814	1.8285	61.89	0.01616
7	176.85	144.7	991.8	1136.5	922.4	69.4	0.2579	1.5582	1.8161	53.56	0.01867
8	182.86	150.8	988.2	1139.0	918.2	70.0	0.2673	1.5380	1.8053	47.27	0.02115
9	188.27	156.2	985.0	1141.1	914.4	70.6	0.2756	1.5202	1.7958	42.36	0.02361
10	193.22	161.1	982.0	1143.1	910.9	71.1	0.2832	1.5042	1.7874	38.38	0.02606
11	197.75	165.7	979.2	1144.9	907.8	71.5	0.2902	1.4895	1.7797	35.10	0.02849
12	201.96	169.9	976.6	1146.5	904.8	71.8	0.2967	1.4760	1.7727	32.36	0.03090

* Courtesy of the Publishers, Longmans, Green & Co. † Interpolated.

PROPERTIES OF SATURATED STEAM — (*Continued*).

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F.	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapor.	Total Entropy.	Specific Volume	Density Weight per Cubic Foot, Pounds.
p	t	q	r	$\lambda = r + q$	ρ	Apu	θ	$\frac{r}{T}$	$\theta + \frac{r}{T}$	S	γ
13	205.87	173.8	974.2	1148.0	902.0	72.2	0.3025	1.4639	1.7664	30.03	0.03330
14	209.55	177.5	971.9	1149.4	899.3	72.6	0.3081	1.4523	1.7604	28.02	0.03569
14.7	212.00	180.0	970.4	1150.4	897.6	72.9	0.3118	1.4447	1.7565	26.79	0.03732
15	213.0	181.0	969.7	1150.7	896.8	72.9	0.3133	1.4416	1.7549	26.27	0.03806
20	228.0	196.1	960.0	1156.2	885.8	74.3	0.3355	1.3965	1.7320	20.08	0.04980
25	240.1	208.4	952.0	1160.4	876.8	75.3	0.3532	1.3604	1.7136	16.30	0.0614
30	250.3	218.8	945.1	1163.9	869.0	76.2	0.3680	1.3311	1.6991	13.74	0.0728
35	259.3	227.9	938.9	1166.8	862.1	76.9	0.3868	1.3060	1.6868	11.89	0.0841
40	267.3	236.1	933.3	1169.4	855.9	77.6	0.3920	1.2841	1.6761	10.49	0.0953
45	274.5	243.4	928.2	1171.6	850.3	78.1	0.4021	1.2644	1.6665	9.39	0.1065
50	281.0	250.1	923.5	1173.6	845.0	78.6	0.4113	1.2468	1.6581	8.51	0.1175
55	287.1	256.3	919.0	1175.4	840.2	78.9	0.4196	1.2309	1.6505	7.78	0.1285
60	292.7	262.1	914.9	1177.0	835.6	79.7	0.4272	1.2160	1.6432	7.17	0.1394
65	298.0	265.7	911.0	1178.5	831.4	79.8	0.4344	1.2034	1.6368	6.65	0.1503
70	302.9	272.6	907.2	1179.8	827.3	80.1	0.4411	1.1896	1.6307	6.20	0.1612
75	307.6	277.4	903.7	1181.8	823.5	80.5	0.4474	1.1778	1.6252	5.81	0.1721
80	312.0	282.0	900.3	1182.3	819.8	80.7	0.4535	1.1665	1.6200	5.47	0.1829
85	316.3	286.3	897.1	1183.4	816.3	81.0	0.4590	1.1561	1.6151	5.16	0.1937
90	320.3	290.5	893.9	1184.4	813.0	81.2	0.4644	1.1461	1.6105	4.89	0.2044
95	324.0	294.5	890.9	1185.4	809.7	81.5	0.4694	1.1367	1.6061	4.65	0.2151
100	327.8	298.3	888.0	1186.3	806.6	81.7	0.4743	1.1277	1.6020	4.429	0.2258

105	331.4	302.0	885.2	1187.2	803.6	81.9	0.4789	1.1191	1.5980	4.230	0.2365
110	334.8	305.5	882.5	1188.0	800.7	82.1	0.4834	1.1108	1.5942	4.047	0.2472
115	338.1	309.0	879.8	1188.8	797.9	82.3	0.4877	1.1030	1.5907	3.880	0.2577
120	341.3	312.3	877.2	1189.6	795.2	82.5	0.4919	1.0954	1.5873	3.726	0.2683
125	344.4	315.5	874.7	1190.3	792.6	82.6	0.4959	1.0880	1.5839	3.583	0.2791
130	347.4	318.6	872.3	1191.0	790.0	82.8	0.4998	1.0809	1.5807	3.452	0.2897
135	350.3	321.7	869.9	1191.6	787.5	82.9	0.5035	1.0742	1.5777	3.331	0.3002
140	353.1	324.6	867.6	1192.2	785.0	83.0	0.5072	1.0675	1.5747	3.219	0.3107
145	355.8	327.4	865.4	1192.8	782.7	83.2	0.5107	1.0612	1.5719	3.112	0.3213
150	358.5	330.2	863.2	1193.4	780.4	83.3	0.5142	1.0550	1.5692	3.012	0.3320
155	361.0	332.9	861.0	1194.0	778.1	83.5	0.5175	1.0489	1.5664	2.920	0.3425
160	363.6	335.6	858.8	1194.5	775.8	83.6	0.5208	1.0431	1.5639	2.834	0.3529
165	366.0	338.2	856.8	1195.0	773.6	83.7	0.5239	1.0376	1.5615	2.753	0.3633
170	368.5	340.7	854.7	1195.4	771.5	83.8	0.5269	1.0321	1.5590	2.675	0.3738
175	370.8	343.2	852.7	1195.9	769.4	83.9	0.5299	1.0268	1.5567	2.602	0.3843
180	373.1	345.6	850.8	1196.4	767.4	84.0	0.5328	1.0215	1.5543	2.533	0.3948
185	375.4	348.0	848.8	1196.8	765.4	84.1	0.5356	1.0164	1.5520	2.468	0.4052
190	377.6	350.4	846.9	1197.3	763.4	84.2	0.5381	1.0114	1.5498	2.406	0.4157
195	379.8	352.7	845.0	1197.7	761.4	84.3	0.5410	1.0066	1.5476	2.346	0.4262
200	381.9	354.9	843.2	1198.1	759.5	84.4	0.5437	1.0019	1.5456	2.290	0.437
205	384.0	357.1	841.4	1198.5	757.6	84.5	0.5463	0.9973	1.5436	2.237	0.447
210	386.0	359.2	839.6	1198.8	755.8	84.5	0.5488	0.9928	1.5416	2.187	0.457
215	388.0	361.4	837.9	1199.2	754.0	84.6	0.5513	0.9885	1.5398	2.138	0.468
220	389.9	363.4	836.2	1199.6	752.3	84.7	0.5538	0.9841	1.5379	2.091	0.478
225	391.9	365.5	834.4	1199.9	750.5	84.7	0.5562	0.9799	1.5361	2.046	0.489
230	393.8	367.5	832.8	1200.2	748.8	84.8	0.5586	0.9758	1.5341	2.004	0.499
240	397.4	371.4	829.5	1200.9	745.4	85.0	0.5633	0.9676	1.5309	1.924	0.520
250	401.1	375.2	826.3	1201.5	742.0	85.1	0.5676	0.9600	1.5276	1.850	0.541
275	409.5	384.2	818.6	1202.8	734.2	85.3	0.5780	0.9419	1.5199	1.686	0.593
300	417.5	392.7	811.3	1204.1	726.8	85.6	0.5878	0.9251	1.5123	1.551	0.645

PROPERTIES OF SATURATED STEAM — (*Continued*).*

Absolute Pressure, Pounds per Square Inch.	Temperature, Degrees F.	Heat of the Liquid.	Heat of Vaporization.	Total Heat.	Heat Equivalent of Internal Work.	Heat Equivalent of External Work.	Entropy of the Liquid.	Entropy of the Vapor.	Total Entropy.	Specific Volume.	Density Weight per Cubic Foot, Pounds.
p	t	q	r	$\lambda = r + q$	ρ	Apu	θ	$\frac{r}{T}$	$\theta + \frac{r}{T}$	S	γ
325	424.9	400.4	801.4	1202.2	717.0	84.7	0.5953	0.9065	1.5018	1.428	0.700
350	431.9	408.0	794.5	1202.5	709.7		0.6036	0.8912	1.4949	1.327	0.753
375	438.5	415.1	787.5	1202.6	702.7		0.6115	0.8768	1.4884	1.239	0.807
400	444.8	422.0	780.6	1202.5	695.9	84.6	0.6190	0.8631	1.4821	1.162	0.860
425	450.7	428.5	773.9	1202.4	689.4		0.6261	0.8501	1.4762	1.121	0.914
450	456.5	434.8	767.4	1202.2	683.1		0.6329	0.8377	1.4706	1.033	0.968
500	467.2	446.6	755.0	1201.7	670.9	84.0	0.6455	0.8146	1.4601	0.928	1.077
550	477.2	458.1	743.3	1200.8	659.2		0.6571	0.7934	1.4506	0.843	1.132
600	486.5	468.0	731.8	1199.8	648.5	83.0	0.6679	0.7735	1.4414	0.770	1.30
650	495.2	477.8	720.9	1198.7	638.0		0.6780	0.7550	1.4330	0.708	1.41
700	503.4	487.1	710.3	1197.4	627.9	82.3	0.6874	0.7376	1.4250	0.656	1.52
750	511.1	495.9	700.0	1195.9	618.2		0.6963	0.7212	1.4175	0.610	1.64
800	518.5	504.3	690.1	1194.4	608.8	81.2	0.7048	0.7056	1.4104	0.570	1.76
850	525.5	512.5	680.4	1192.8	599.7		0.7128	0.6907	1.4035	0.534	1.87
900	532.3	520.3	670.8	1191.9	590.8	80.1	0.7205	0.6764	1.3969	0.502	1.99
1000	544.9	535.2	652.4	1187.6	573.6	78.6	0.7349	0.6496	1.3845	0.447	2.24
1100	556.6	549.1	634.7	1183.8	557.3	77.6	0.7482	0.6247	1.3729	0.403	2.48
1200	567.7	562.3	617.6	1179.7	541.8	75.6	0.7607	0.6015	1.3622	0.364	2.74
1500	596.4	599.0	566.6	1167.0	495.0	73.5	0.7970	0.5360	0.1334	0.281	3.56
2000	635.9	657.0	480.0	1138.0	418.0	62.0	......	0.4390	......	0.194	5.18
3200	706.3	921.0	0.0	921.0	0.0	0.0	......	......	0.0	0.048	20.90

* G. A. Goodenough.

Total Entropy of Superheated Steam.

$$N = n + n_s + \theta = \frac{r}{T} + C_m \log_e \frac{T_s}{T_v} + \theta. \tag{342}$$

Using Knoblauch and Jakob's values for the specific heat of superheated steam, Goodenough gives the following rule for calculating the total entropy of superheated steam

$$N_s = 0.73683 \log T_s + 0.000126\ T_s - \frac{11791.5}{T_s^2} - 0.2535 \log p$$
$$- \frac{C_4 p\ (1 + 0.0342\ p)}{T_s^3} - 0.08085. \tag{343}$$

$\log C_4 = 10.69464$.

Tables 136 and 137 are abridged from Marks and Davis' "Steam Tables and Diagrams."

386. Mollier Diagram. — Steam tables are often accompanied by graphical charts that may be used to great advantage in the solution of thermodynamic problems. Fig. 686 gives a skeleton outline of the total heat-entropy diagram and Fig. 687 a reduced copy of the complete chart. The first conception of the heat-entropy chart is due to Dr. R. Mollier of Dresden, hence the name, Mollier Diagram.

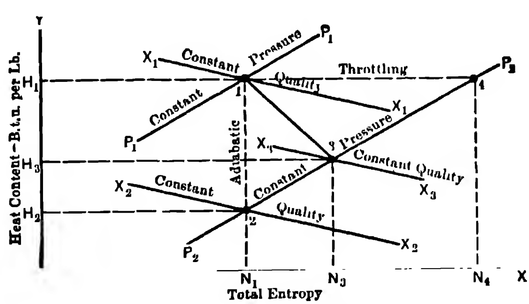


FIG. 686. Mollier Diagram — Skeleton Outline.

Referring to Fig. 686, abscissas represent total entropy and ordinates represent B.t.u. per lb. Vertical lines then indicate constant entropy and horizontal lines constant heat content. P_1P_1 and P_2P_2 represent lines of constant pressure and X_1X_1 and X_2X_2 lines of constant quality. Evidently any point in the chart represents a fixed condition of heat content, pressure, quality, and entropy as determined by its location with respect to the different lines. Thus, point 1 represents a pressure P_1 as determined by the numerical value of line P_1P_1, quality X_1 by its location on line X_1X_1, entropy N_1 by its projection on the X axis and heat content H_1 by its projection on the Y axis.

TABLE 137.

PROPERTIES OF SUPERHEATED STEAM.

Reproduced by Permission from Marks and Davis' "Steam Tables and Diagrams."

Pressure, Pounds Absolute		Saturated Steam.	Degrees of Superheat. 50	100	150	200	250	300		Pressure, Pounds Absolute.
	t	162 3	212 3	262 3	312 3	362 3	412 3	462.3	*t*	
5	*v*	73 3	79 7	85 7	91 8	97.8	103.8	109.8	*v*	5
	h	1130.5	1153 5	1176 4	1199 5	1222 5	1245.6	1268.7	*h*	
	t	193.2	243 2	293.2	343 2	393 2	443 2	493.2	*t*	
10	*v*	38 4	41.5	44 6	47.7	50.7	53.7	56.7	*v*	10
	h	1143 1	1166.3	1189.5	1212 7	1236.0	1259 3	1282 5	*h*	
	t	213.0	263 0	313 0	363 0	413.0	463.0	513.0	*t*	
15	*v*	26 27	28.40	30.46	32 50	34.53	36 56	38.58	*v*	15
	h	1150 7	1174 2	1197.6	1221.0	1244.4	1267 7	1291.1	*h*	
	t	228.0	278.0	328.0	378.0	428 0	478.0	528.0	*t*	
20	*v*	20.08	21.69	23.25	24 80	26.33	27 85	29 37	*v*	20
	h	1156 2	1179.9	1203 5	1227.1	1250.6	1274 1	1297.6	*h*	
	t	240 1	290.1	340.1	390 1	440 1	490 1	540 1	*t*	
25	*v*	16 30	17 60	18.86	20 10	21.32	22 55	23 77	*v*	25
	h	1160 4	1184.4	1208.2	1231 9	1255.6	1279.2	1302.8	*h*	
	t	250 4	300.4	350 4	400.4	450.4	500.4	550 4	*t*	
30	*v*	13.74	14.83	15.89	16 93	17.97	18.99	20 00	*v*	30
	h	1163.9	1188.1	1212.1	1236.0	1259.7	1283.4	1307.1	*h*	
	t	259 3	309.3	359.3	409.3	459.3	509.3	559.3	*t*	
35	*v*	11.89	12 85	13.75	14 65	15.51	16.42	17.30	*v*	35
	h	1166.8	1191.3	1215 4	1239.4	1263.3	1287.1	1310.8	*h*	
	t	267.3	317 3	367.2	417.3	467.3	517.3	567.3	*t*	
40	*v*	10.49	11 33	12.13	12 93	13.70	14.48	15.25	*v*	40
	h	1169.4	1194.0	1218 4	1242.4	1266.4	1290.3	1314.1	*h*	
	t	274 5	324.5	374 5	424.5	474.5	524.5	574.5	*t*	
45	*v*	9.39	10 14	10 86	11.57	12 27	12 96	13.65	*v*	45
	h	1171 6	1196.6	1221.0	1245.2	1269.3	1293.2	1317 0	*h*	
	t	281.0	331 0	381.0	431 0	481 0	531 0	581.0	*t*	
50	*v*	8 51	9.19	9 84	10 48	11.11	11.74	12 36	*v*	50
	h	1173 6	1198 8	1223 4	1247 7	1271 8	1295.8	1319 7	*h*	
	t	287.1	337.1	387.1	437.1	487.1	537.1	587.1	*t*	
55	*v*	7.78	8.40	9.00	9 59	10.16	10.73	11 30	*v*	55
	h	1175.4	1200 8	1225.6	1250.0	1274 2	1298 1	1322.0	*h*	
	t	292.7	342.7	392 7	442 7	492 7	542 7	592 7	*t*	
60	*v*	7.17	7.75	8 30	8.84	9 36	9.89	10 41	*v*	60
	h	1177.0	1202.6	1227.6	1252.1	1276.4	1300.4	1324.3	*h*	
	t	298.0	348 0	398 0	448.0	498.0	548 0	598 0	*t*	
65	*v*	6.65	7.20	7.70	8 20	8 69	9.17	9 65	*v*	65
	h	1178 5	1204.4	1229.5	1254.0	1278.4	1302 4	1326 4	*h*	
	t	302.9	352 9	402.9	452.9	502.9	552.9	602.9	*t*	
70	*v*	6.20	6 71	7.18	7.65	8 11	8 56	9 01	*v*	70
	h	1179 8	1205.9	1231.2	1255.8	1280.2	1304.3	1328 3	*h*	
	t	307 6	357 6	407 6	457.6	507.6	557.6	607.6	*t*	
75	*v*	5.81	6 28	6 73	7.17	7.60	8.02	8 44	*v*	75
	h	1181 1	1207.5	1232.8	1257.5	1282.0	1306 1	1330.1	*h*	
	t	312 0	362 0	412.0	462.0	512.0	562 0	612 0	*t*	
80	*v*	5.47	5 92	6.34	6 75	7.17	7.56	7.95	*v*	80
	h	1182.3	1208.8	1234.3	1259 0	1283 6	1307.8	1331 9	*h*	
	t	316.3	366 3	416 3	466.3	516 3	566.3	616.3	*t*	
85	*v*	5.16	5.59	5.99	6.38	6.76	7.14	7.51	*v*	85
	h	1183 4	1210.2	1235.8	1260.6	1285.2	1309.4	1333 5	*h*	

t = Temperature, deg fahr.
v = Specific volume, in cubic feet, per pound.
h = Total heat from water at 32 degrees. B.t.u.

TABLE 137. — *Continued*

Pressure, Pounds Absolute.		Saturated Steam.	Degrees of Superheat. 50	100	150	200	250	300		Pressure, Pounds Absolute.
	t	320.3	370.3	420.3	470.3	520.3	570.3	620.3	*t*	
90	*v*	4.89	5.29	5.67	6.04	6.40	6.76	7.11	*v*	90
	h	1184.4	1211.4	1237.2	1262.0	1286.6	1310.8	1334.9	*h*	
	t	324.1	374.1	424.1	474.1	524.1	574.1	624.1	*t*	
95	*v*	4.65	5.03	5.39	5.74	6.09	6.43	6.76	*v*	95
	h	1185.4	1212.6	1238.4	1263.4	1288.1	1312.3	1336.4	*h*	
	t	327.8	377.8	427.8	477.8	527.8	577.8	627.8	*t*	
100	*v*	4.43	4.79	5.11	5.47	5.80	6.12	6.41	*v*	100
	h	1186.3	1213.8	1239.7	1264.7	1289.4	1313.6	1337.8	*h*	
	t	331.4	381.4	431.4	481.4	531.4	581.4	631.4	*t*	
105	*v*	4.23	4.58	4.91	5.23	5.54	5.85	6.15	*v*	105
	h	1187.2	1214.9	1240.8	1265.9	1290.6	1314.9	1339.1	*h*	
	t	334.8	384.8	434.8	484.8	534.8	584.8	634.8	*t*	
110	*v*	4.05	4.38	4.70	5.01	5.31	5.61	5.90	*v*	110
	h	1188.0	1215.9	1242.0	1267.1	1291.9	1316.2	1340.4	*h*	
	t	338.1	388.1	438.1	488.1	538.1	588.1	638.1	*t*	
115	*v*	3.88	4.20	4.51	4.81	5.09	5.38	5.66	*v*	115
	h	1188.8	1216.9	1243.1	1268.2	1293.0	1317.3	1341.5	*h*	
	t	341.3	391.3	441.3	491.3	541.3	591.3	641.3	*t*	
120	*v*	3.73	4.04	4.33	4.62	4.89	5.17	5.44	*v*	120
	h	1189.6	1217.9	1244.1	1269.3	1294.1	1318.4	1342.7	*h*	
	t	344.4	394.4	444.4	494.4	544.4	594.4	644.4	*t*	
125	*v*	3.58	3.88	4.17	4.45	4.71	4.97	5.23	*v*	125
	h	1190.3	1218.8	1245.1	1270.4	1295.2	1319.5	1343.8	*h*	
	t	347.4	397.4	447.4	497.4	547.4	597.4	647.4	*t*	
130	*v*	3.45	3.74	4.02	4.28	4.51	4.80	5.05	*v*	130
	h	1191.0	1219.7	1246.1	1271.4	1296.2	1320.6	1344.9	*h*	
	t	350.3	400.3	450.3	500.3	550.3	600.3	650.3	*t*	
135	*v*	3.33	3.61	3.88	4.14	4.38	4.63	4.87	*v*	135
	h	1191.6	1220.6	1247.0	1272.3	1297.2	1321.6	1345.9	*h*	
	t	353.1	403.1	453.1	503.1	553.1	603.1	653.1	*t*	
140	*v*	3.22	3.49	3.75	4.00	4.24	4.48	4.71	*v*	140
	h	1192.2	1221.4	1248.0	1273.3	1298.2	1322.6	1346.9	*h*	
	t	355.8	405.8	455.8	505.8	555.8	605.8	655.8	*t*	
145	*v*	3.12	3.38	3.63	3.87	4.10	4.33	4.56	*v*	145
	h	1192.8	1222.2	1248.8	1274.2	1299.1	1323.6	1347.9	*h*	
	t	358.5	408.5	458.5	508.5	558.5	608.5	658.5	*t*	
150	*v*	3.01	3.27	3.51	3.75	3.97	4.19	4.41	*v*	150
	h	1193.4	1223.0	1249.6	1275.1	1300.0	1324.5	1348.8	*h*	
	t	361.0	411.0	461.0	511.0	561.0	611.0	661.0	*t*	
155	*v*	2.92	3.17	3.41	3.63	3.85	4.06	4.28	*v*	155
	h	1194.0	1223.6	1250.5	1276.0	1300.8	1325.3	1349.7	*h*	
	t	363.6	413.6	463.6	513.6	563.6	613.6	663.6	*t*	
160	*v*	2.83	3.07	3.30	3.53	3.74	3.95	4.15	*v*	160
	h	1194.5	1224.5	1251.3	1276.8	1301.7	1326.2	1350.6	*h*	
	t	366.0	416.0	466.0	516.0	566.0	616.0	666.0	*t*	
165	*v*	2.75	2.99	3.21	3.43	3.64	3.84	4.04	*v*	165
	h	1195.0	1225.2	1252.0	1277.6	1302.5	1327.1	1351.5	*h*	
	t	368.5	418.5	468.5	518.5	568.5	618.5	668.5	*t*	
170	*v*	2.68	2.91	3.12	3.34	3.54	3.73	3.92	*v*	170
	h	1195.4	1225.9	1252.8	1278.4	1303.3	1327.9	1352.3	*h*	

t = Temperature, deg. fahr.
v = Specific volume, in cubic feet, per pound.
h = Total heat from water at 32 degrees, B.t.u.

TABLE 137. — *Continued*

Pressure, Pounds Absolute.		Saturated Steam.	Degrees of Superheat. 50	100	150	200	250	300		Pressure, Pounds Absolute.
	t	370.8	420.8	470.8	520.8	570.8	620.8	670.8	*t*	
175	*v*	2.60	2.83	3.04	3.24	3.44	3.63	3.82	*v*	175
	h	1195.9	1226.6	1253.6	1279.1	1304.1	1328.7	1353.2	*h*	
	t	373.1	423.1	473.1	523.1	573.1	623.1	673.1	*t*	
180	*v*	2.53	2.75	2.96	3.16	3.35	3.54	3.72	*v*	180
	h	1196.4	1227.2	1254.3	1279.9	1304.8	1329.5	1353.9	*h*	
	t	375.4	425.4	475.4	525.4	575.4	625.4	675.4	*t*	
185	*v*	2.47	2.68	2.89	3.08	3.27	3.45	3.63	*v*	185
	h	1196.8	1227.9	1255.0	1280.6	1305.6	1330.2	1354.7	*h*	
	t	377.6	427.6	477.6	527.6	577.6	627.6	677.6	*t*	
190	*v*	2.41	2.62	2.81	3.00	3.19	3.37	3.55	*v*	190
	h	1197.3	1228.6	1255.7	1281.3	1306.3	1330.9	1355.5	*h*	
	t	379.8	429.8	479.8	529.8	579.8	629.8	679.8	*t*	
195	*v*	2.35	2.55	2.75	2.93	3.11	3.29	3.46	*v*	195
	h	1197.7	1229.2	1256.4	1282.0	1307.0	1331.6	1356.2	*h*	
	t	381.9	431.9	481.9	531.9	581.9	631.9	681.9	*t*	
200	*v*	2.29	2.49	2.68	2.86	3.04	3.21	3.38	*v*	200
	h	1198.1	1229.8	1257.1	1282.6	1307.7	1332.4	1357.0	*h*	
	t	384.0	434.0	484.0	534.0	584.0	634.0	684.0	*t*	
205	*v*	2.24	2.44	2.62	2.80	2.97	3.14	3.30	*v*	205
	h	1198.5	1230.4	1257.7	1283.3	1308.3	1333.0	1357.7	*h*	
	t	386.0	436.0	486.0	536.0	586.0	636.0	686.0	*t*	
210	*v*	2.19	2.38	2.56	2.74	2.91	3.07	3.23	*v*	210
	h	1198.8	1231.0	1258.4	1284.0	1309.0	1333.7	1358.4	*h*	
	t	388.0	438.0	488.0	538.0	588.0	638.0	688.0	*t*	
215	*v*	2.14	2.33	2.51	2.68	2.84	3.00	3.16	*v*	215
	h	1199.2	1231.6	1259.0	1284.6	1309.7	1334.4	1359.1	*h*	
	t	389.9	439.9	489.9	539.9	589.9	639.9	689.9	*t*	
220	*v*	2.09	2.28	2.45	2.62	2.78	2.94	3.10	*v*	220
	h	1199.6	1232.2	1259.6	1285.2	1310.3	1335.1	1359.8	*h*	
	t	391.9	441.9	491.9	541.9	591.9	641.9	691.9	*t*	
225	*v*	2.05	2.23	2.40	2.57	2.72	2.88	3.03	*v*	225
	h	1199.9	1232.7	1260.2	1285.9	1310.9	1335.7	1360.3	*h*	
	t	393.8	443.8	493.8	543.8	593.8	643.8	693.8	*t*	
230	*v*	2.00	2.18	2.35	2.51	2.67	2.82	2.97	*v*	230
	h	1200.2	1233.2	1260.7	1286.5	1311.6	1336.3	1361.0	*h*	
	t	395.6	445.6	495.6	545.6	595.6	645.6	695.6	*t*	
235	*v*	1.96	2.14	2.30	2.46	2.62	2.77	2.91	*v*	235
	h	1200.6	1233.8	1261.4	1287.1	1312.2	1337.0	1361.7	*h*	
	t	397.4	447.4	497.4	547.4	597.4	647.4	697.4	*t*	
240	*v*	1.92	2.09	2.26	2.42	2.57	2.71	2.85	*v*	240
	h	1200.9	1234.3	1261.9	1287.6	1312.8	1337.6	1362.3	*h*	
	t	399.3	449.3	499.3	549.3	599.3	649.3	699.3	*t*	
245	*v*	1.89	2.05	2.22	2.37	2.52	2.66	2.80	*v*	245
	h	1201.2	1234.8	1262.5	1288.2	1313.3	1338.2	1362.9	*h*	
	t	401.0	451.0	501.0	551.0	601.0	651.0	701.0	*t*	
250	*v*	1.85	2.02	2.17	2.33	2.47	2.61	2.75	*v*	250
	h	1201.5	1235.4	1263.0	1288.8	1313.9	1338.8	1363.5	*h*	
	t	402.8	452.8	502.8	552.8	602.8	652.8	702.8	*t*	
255	*v*	1.81	1.98	2.14	2.28	2.43	2.56	2.70	*v*	255
	h	1201.8	1235.9	1263.6	1289.3	1314.5	1339.3	1364.1	*h*	

t = Temperature, deg. fahr.
v = Specific volume, in cubic feet, per pound.
h = Total heat from water at 32 degrees, B.t.u.

In addition to the Mollier diagram, the Marks and Davis tables include a total heat-pressure diagram which is of great assistance in the solution of problems involving ratios of expansion.

The Ellenwood Charts (John Wiley & Sons, Publishers) have a much wider field of application than the diagrams mentioned above and afford a simple and accurate means of solving practically all thermodynamic problems involving the use of the properties of steam.

Steam Table Research · Power, Apr. 5, 1924, p. 200; Apr. 12, 1924, p 246; Apr. 19, 1924, p. 284; Apr. 26, 1924, p 329

Experiments on the Generation of "Critical" Steam Power, Oct. 28, 1924, p. 693.

CHAPTER XXII—Supplementary

ELEMENTARY THERMODYNAMICS — CHANGE OF STATE

387. General. — The laws governing the transformation of steam from one state to another form the basis of practically all thermodynamic analyses of the steam engine and turbine. The more common and important changes are

(1) Isobaric or equal pressure.
(2) Isovolumic or equal volume.
(3) Isothermal or equal temperature.
(4) Constant heat content.
(5) Adiabatic or no external heat exchange.
(6) Polytropic.

388. Isobaric or Equal-Pressure Change. *Saturated Vapor.* — Since the temperature of wet or saturated steam is dependent on the pressures only, a constant-pressure change of such material must also be a constant-temperature one. Denoting the initial and final properties by subscripts 1 and 2 respectively:

$$\text{Initial volume } v_1 = x_1 s_1 + (1 - x_1)\,\sigma_1 = x_1 u_1 + \sigma_1. \tag{344}$$

$$\text{Final volume } v_2 = x_2 s_1 + (1 - x_2)\,\sigma_1 = x_2 u_1 + \sigma_1. \tag{345}$$

$$\text{Change of volume } v_2 - v_1 = u_1\,(x_2 - x_1). \tag{346}$$

$$\text{External work } W = P_1\,(v_2 - v_1) \quad P_1 u_1\,(x_2 - x_1). \tag{347}$$

$$\text{Change of energy } = \frac{\rho_1}{A}\,(x_2 - x_1). \tag{348}$$

$$\text{Heat absorbed } = r_1\,(x_2 - x_1). \tag{349}$$

Notations:

$A = \frac{1}{778}$. p = lb. per sq. in. abs.
P = lb. per sq. ft. abs. x = quality of wet steam.
s = specific volume of dry steam, cu. ft. per lb.
v = specific volume of vapor, cu. ft. per lb.
σ = specific volume of water, cu. ft. per lb.
u = increase in volume during evaporation, cu. ft.
t = deg. fahr. above zero. T = deg. fahr. abs.
c_m = mean specific heat of water.
C = mean specific heat of superheated steam.
H = heat content above 32 deg. fahr., B.t.u. per lb.
λ = total heat of dry steam, B.t.u. per lb.
r = latent heat of vaporization, B.t.u. per lb.
ρ = internal latent heat, B.t.u. per lb.
q = heat of liquid, B.t.u. per lb.
θ = entropy of the liquid.
n = entropy of the vapor.
N = total entropy.
Prime marks indicate superheat.
Subscripts *1*, *2*, *w*, *s* indicate, respectively, initial condition, final condition, wet steam, and superheated steam.

Example 95. — At a pressure of 115 lb. per sq. in. absolute, the volume of one pound of vapor and liquid is increased 1 cu. ft. Required the change of quality, external work, increase of energy and heat absorbed.

Solution. — From steam tables $s_1 = 3.88$; $\sigma_1 = 0.0179$; $\rho_1 = 797.9$; $r_1 = 879.8$.

$$\text{Change of quality} = x_2 - x_1 = \frac{v_2 - v_1}{s_1 - \sigma_1} = \frac{1}{3.88 - 0.0179} = 0.259.$$

$$\text{External work} = P_1 (v_2 - v_1) = 144 \times 115 \times 1 = 16{,}560 \text{ ft. lb.}$$

$$\text{Change of energy} = \frac{\rho}{A} (x_2 - x_1) = 797.9 \times 778 \times 0.259$$

$$= 160{,}778 \text{ ft. lb.}$$

$$\text{Heat absorbed} = r_1 (x_2 - x_1) = 879.8 \times 0.259 = 227.79 \text{ B.t.u.}$$

Superheated Steam. — Let superheated steam change state at constant pressure p_1 from an initial temperature t_1 to a final temperature t_2.

Change of volume $= v_2' - v_1'$. The values of v' corresponding to pressure p_1 and temperatures t_1 and t_2 may be taken directly from steam tables or they may be calculated from equation (308). They may be approximated from equations (311) and (312).

$$\text{External work} = P_1 (v_2' - v_1'). \tag{350}$$

$$\text{Change of energy} = \left(\frac{H_2'}{A} - P_1 v_2'\right) - \left(\frac{H_1'}{A} - P_1 v_1'\right). \tag{351}$$

$$= \frac{H_2' - H_1'}{A} - P_1 (v_2' - v_1'). \tag{352}$$

$$\text{Heat absorbed} = H_2' - H_1'. \tag{353}$$

$$\text{Change of entropy} = N_2' - N_1'. \tag{354}$$

Example 96. — Using the data in the preceding example, determine the various quantities, if the initial degree of superheat is 100 deg. fahr.

Solution. — From superheated steam tables for $p_1 = 115$ and $t_1 = 438.1$ $(= 338.1 + 100)$ we find: $v_1' = 4.51$; $H_1' = 1243.1$; $N_1' = 1.6549$.

For $p_2 = p_1 = 115$ and $v_2' = (4.51 + 1) = 5.51$ we find by interpolation $H_2' = 1328.5$; $N_2' = 1.7419$; $t_2' = 621.3$.

$$\text{Increase of superheat} = t_2' - t_1'$$
$$= 621.3 - 438.1 = 183.2 \text{ deg. fahr.}$$

$$\text{External work} = P_1 (v_2' - v_1')$$
$$= 144 \times 115 \times 1 = 16{,}560 \text{ ft. lb.}$$

$$\text{Increase of energy} = \frac{H_2' - H_1'}{A} - P_1 (v_2' - v_1')$$
$$= (1328.5 - 1243.1)778 - 16{,}560$$
$$= 49{,}881 \text{ ft. lb.}$$

$$\text{Increase of entropy} = N_1' - N_2'$$
$$= 1.7419 - 1.6549 = 0.087.$$

$$\text{Heat absorbed} = H_2' - H_1'$$
$$= 1328.5 - 1243.1 = 85.4 \text{ B.t.u.}$$

389. Isovolumic or Equal Volume Change. *Saturated Steam.* — Since the volumes s_1 and s_2 are equal

$$s_1 = s_2 \text{ or } x_1u_1 + \sigma_1 = x_2u_2 + \sigma_2. \tag{355}$$

$$\text{External work} = 0. \tag{356}$$

$$\text{Heat absorbed} = x_1\rho_1 + q_1 - (x_2\rho_2 + q_2). \tag{357}$$

Example 97. — A pound of mixture of vapor and liquid at 115 lb. per sq. in. absolute and quality 0.9 is cooled at constant volume to a pressure of 1 lb. per sq. in. absolute. Required the various properties at the final condition and the heat taken from the mixture.

Solution. — From steam tables:

$$p_1 = 115,\ s_1 = 3.88,\ \sigma_1 = 0.0179,$$
$$\rho_1 = 797.9,\ q_1 = 309,\ n_1 = 1.103,\ \theta_1 = 0.4877,$$
$$p_2 = 1,\ s_2 = 333,\ \sigma_2 = 0.0161,\ \rho_2 = 972.9,$$
$$q_2 = 69.8,\ n_2 = 1.8427,\ \theta_2 = 0.1327,$$

$$\text{Final quality } x_2 = \frac{x_1u_1 + \sigma_1 - \sigma_2}{u_2}$$
$$= \frac{0.9\,(3.88 - 0.0179) + 0.0179 - 0.0161}{333 - 0.0161}$$
$$= 0.0105.$$

$$\text{Heat removed} = x_1\rho_1 + q_1 - (x_2\rho_2 + q_2)$$
$$= 0.9 \times 797.9 + 309 - (0.0105 \times 972.9 + 69.8)$$
$$= 947 \text{ B.t.u.}$$

$$\text{Initial entropy } N_1 = x_1n_1 + \theta_1$$
$$= 0.9 \times 1.103 + 0.4877 = 1.4804.$$

$$\text{Final entropy } N_2 = x_2n_2 + \theta_2$$
$$= 0.0105 \times 1.8427 + 0.1327 = 0.1520.$$

Superheated Steam. — Since the final volume is equal to the initial, and both pressures and the initial temperature are known, the final temperature may be calculated from equation (308) or it may be taken directly or interpolated from the steam tables.

Example 98. — Using the data in the preceding problem determine the various factors if the initial degree of superheat is 100 deg. fahr.

Solution. — From steam tables for $p_1 = 115$ and $t_1 = 338.1 + 100 = 438.1$ we find: $v_1' = 4.51$, $H_1' = 1243.1$, $N_1' = 1.6549$.

s for 1 lb. per sq. in. absolute pressure = 333 cu. ft. but the given volume is 4.51 cu. ft. Therefore the steam is wet at the final condition.

From steam tables for $p_2 = 1$ we find:

$$\rho_2 = 972.9,\ q_2 = 69.8,\ n_2 = 1.8427,\ \theta_2 = 0.1327.$$

Since the volumes are equal

$$v_1' = v_2$$
$$= x_2u_2 + \sigma_2.$$

$$\text{Final quality } x_2 = \frac{v_1' - \sigma_2}{u}$$

$$= \frac{4.51 - 0.0161}{333 - 0.0161} = 0.0135.$$

$$\text{Heat removed} = H_1' - AP_1v_1' - (x_2\rho_2 + q_2)$$

$$= 1243.1 - \frac{144 \times 115}{778} \times 4.51 - (0.0135 \times 972.9 + 69.8)$$

$$= 1065 \text{ B.t.u.}$$

Initial entropy (from steam tables) $N_1' = 1.6549$.
Final entropy $N_2 = x_2n_2 + \theta_2$
$= 0.0135 \times 1.8427 + 0.1327 = 0.1575$.

390. Isothermal or Equal Temperature Change. *Saturated Vapor.* — Since the temperature of wet or saturated steam is dependent solely upon the pressure, an isothermal change is also isobaric, and the data in paragraph (388) are applicable to this change.

Superheated Steam. — The properties at initial and final conditions may be calculated from equations of the properties of superheated steam or they may be taken directly from steam tables or charts. If wet or saturated steam expands isothermally into the superheated state the pressure must drop in order to maintain constant temperature. The relation between pressure, volume and temperature for the superheated state is given in equation (308).

Example 99. — One pound of steam at initial pressure 115 lb. per sq. in. absolute and superheat 100 deg. fahr. is expanded isothermally to a pressure of 1 lb. per sq. in. absolute. Required the various properties at the final pressure, the heat absorbed during expansion and the external work done.

Solution. — From superheated steam tables for $p_1 = 115$ and $t_1' = 338.1 + 100 = 438.1$ we find: $v_1' = 4.51$, $H_1' = 1243.1$, $N_1' = 1.6549$.
For $p_2 = 1$ and $t_2' = 438.1$, $v_2' = 535$, $H_2' = 1258.3$, $N_2' = 2.1888$.
Final quality $t_2' - t_2 = 438.1 - 101.8 = 336.3$ deg. superheat.

$$\text{Heat added during expansion} = T_2' \, (N_2' - N_1')$$

$$= 898 \, (2.1888 - 1.6541)$$

$$= 481 \text{ B.t.u.}$$

(Note that the heat added is not equal to the difference in total heats, since the isothermal is not a constant-pressure line.)

$$\text{External work} = \int_1^2 P \, dv. \tag{358}$$

Since the temperature is constant dv may be obtained by differentiating equation (308). Substituting this value of dv in equation (358) and integrating we have,

$$\text{External work} = 85.63\ T_s \log_e \frac{p_1}{p_2} + 2.46\,(p_1^{\frac{1}{4}} - p_2^{\frac{1}{4}})\,\frac{C}{T^4} \qquad (359)$$

$$= 85.63 \times 898 \log_e \frac{115}{1} + 2.46\,(115^{\frac{1}{4}} - 1^{\frac{1}{4}})\,\frac{C}{898^4}$$

$$= 366{,}000 \text{ ft. lb. (approx.)}$$

$$(\log C = 10.8250.)$$

391. Constant Heat Content. — Expansion from one pressure to a lower one with constant heat content is exemplified in throttling or wire drawing. The energy utilized in imparting velocity to the fluid is all returned to the fluid at the lower pressure when the velocity is brought to zero and there are no radiation losses.

For steam wet throughout expansion

$$x_1 r_1 + q_1 = x_2 r_2 + q_2. \qquad (360)$$

For steam initially wet but dry at the lower pressure

$$x_1 r_1 + q_1 = \lambda_2. \qquad (361)$$

For steam initially wet but superheated at the lower pressure

$$x_1 r_1 + q_1 = \lambda_2 + C_m t_2' = H_2'. \qquad (362)$$

For steam initially dry

$$\lambda_1 = \lambda_2 + C_m t_2' = H_2'. \qquad (363)$$

For steam initially superheated

$$H_1' = H_2'. \qquad (364)$$

Loss of available energy due to throttling or wire drawing

$$\text{Loss B.t.u. per lb.} = T_2\,(N_2 - N_1). \qquad (365)$$

Example 100. — One pound of steam at an initial pressure of 115 lb. per sq. in. absolute is expanded through a throttling calorimeter to a pressure of 16 lb. per sq. in. absolute. If the temperature of the steam at the lower pressure is 256.3 deg. fahr. required the initial quality of the steam.

Solution. — From saturated steam tables:

$$p_1 = 115,\ r_1 = 879.8,\ q_1 = 309,\ N_1 = 1.5907.$$

From superheated steam tables for $p_2 = 16$ and $t_2' = 256.3$ we find:

$$H_2 = 1170.8,\ N_2 = 1.7765,\ t_2 \text{ (sat.)} = 216.3,$$

$$x_1 r_1 + q_1 = H_2,$$

$$879.8\, x_1 + 309 = 1170.8, \qquad x_1 = 0.98.$$

Mollier diagram analysis, Fig. 687: From intersection of constant superheat line $t_2' = 40$ $(= 256.3 - 216.3)$ and constant pressure line

$p_2 = 16$ trace horizontally to constant pressure line $p_1 = 115$ and read from its intersection with the constant quality line, $x_1 = 0.98$.

$$\text{Decrease of available energy} = T_2 (N_2 - N_1) \tag{366}$$
$$= (216.3 + 460)(1.7765 - 1.5907)$$
$$= 125.6 \text{ B.t.u.}$$

392. Adiabatic Change of State. — Since in an adiabatic change there is no heat added to or abstracted from the fluid, the entropy remains constant.

Steam wet throughout change of state

$$N_1 = N_2. \tag{366a}$$

$$x_1 n_1 + \theta_1 = x_2 n_2 + \theta_2. \tag{367}$$

$$\frac{x_1 r_1}{T_1} + \theta_1 = \frac{x_2 r_2}{T_2} + \theta_2. \tag{368}$$

For water only, $x = 0$; for dry steam, $x = 1$.

Steam initially superheated but finally wet

$$N_1' = N_2. \tag{369}$$

$$N_1 + n_s = x_2 n_2 + \theta_2. \tag{370}$$

Steam superheated throughout change of state

$$N_1' = N_2', \tag{371}$$

$$N_1 + n_s = N_2 + n_{s(2)}, \tag{372}$$

$$\frac{r_1}{T_1} + \theta_1 + C_m \log_e \frac{T_s}{T_v} = \frac{r_2}{T_2} + \theta_2 + \left. C_m \log_e \frac{T_s}{T_v} \right]_2. \tag{373}$$

Final Quality. Saturated Steam. — This quantity may be calculated directly from equations (366a) and (367).

$$x_2 = \frac{N_1 - \theta_2}{n_2} \tag{374}$$

$$= \left(\frac{x_1 r_1}{T_1} + \theta_1 - \theta_2 \right) \frac{T_2}{r_2}. \tag{375}$$

If water only is present at the beginning of expansion, substitute $N_1 = \theta_1$ in equation (374).

For initial qualities of $x_1 = 0.50$ (approx.) or greater the final quality x_2 decreases as the expansion progresses, and for initial qualities of $x_1 = 0.50$ (approx.) or less the final quality increases. For initial quality $x_1 = 0.50$ the final quality x_2 remains practically constant.

The final volume may be calculated as follows:

$$\text{Wet steam, } v_2 = x_2 u_2 + \sigma_2, \tag{376}$$

x_2 as calculated from equations (367) and (370),

$$\text{Dry steam, } v_2 = s_2. \tag{377}$$

Superheated Steam. — For superheat at the end of expansion the calculations involved in equation (373) are too cumbersome and unwieldy and the Mollier diagram may be used to advantage.

Volume Change. — Superheated steam: the final volume v_2' may be calculated from equation (308) by substituting for p the final pressure, and for T_s the final temperature as calculated from equation (373). The final volume, however, may be taken directly from the pressure-entropy chart.

External Work. — Since the heat added or subtracted is zero, the external work is equal to the change of intrinsic energy, or in general

$$W = \frac{1}{A}\left[(H_1 - AP_1v_1) - (H_2 - AP_2v_2)\right]. \tag{378}$$

Steam initially wet

$$W = \frac{1}{A}\left[(x_1\rho_1 + q_1) - (x_2\rho_2 + q_2)\right]. \tag{379}$$

Steam initially dry, substitute $x_1 = 1$.

Steam initially superheated but wet at end of expansion

$$W = \frac{1}{A}\left[(H_1' - AP_1v_1') - (x_2\rho_2 + q_2)\right]. \tag{380}$$

Steam initially superheated but dry at end of expansion substitute $x_2 = 1$.

Steam superheated throughout expansion

$$W = \frac{1}{A}\left[(H_1' - AP_1v_1') - (H_2' - AP_2v_2')\right]. \tag{381}$$

Heat absorbed $= H_1 - H_2$.

Steam initially wet

$$H_1 - H_2 = (x_1r_1 + q_1) - (x_2r_2 + q_2). \tag{382}$$

x_2 as calculated from equation (374).

Steam initially dry, substitute $x_1 = 1$.

Steam initially superheated but wet at end of expansion

$$H_1' - H_2 = H_1' - (x_2r_2 + q_2). \tag{383}$$

Steam superheated throughout expansion, heat absorbed

$$H_1' - H_2'. \tag{384}$$

Example 101. — One pound of steam at initial pressure 115 lb. per sq. in. absolute and superheat 100 deg. fahr. expands adiabatically to

1 lb. per sq. in. absolute. Required the various quantities at the final condition.

Solution. — From superheated steam tables for $p_1 = 115$ and $t_1' = 438.1 = (338.1 + 100)$ we find: $H_1' = 1243$, $v_1' = 4.51$, $N_1' = 1.6549$.

From saturated steam tables: $p_2 = 1$, $s = 333$, $q_2 = 69.8$, $H_2 = 1104.4$, $r_2 = 1034.6$, $\rho_2 = 972.9$, $n_2 = 1.8427$, $\theta_2 = 0.1327$, $\sigma_2 = .0016$.

Final quality:
$$x_2 = \frac{N_1' - \theta_2}{n_2}$$
$$= \frac{1.6549 - 0.1327}{1.8427} = 0.826.$$

Mollier diagram analysis, Fig. 687: Trace the intersection of $p_1 = 115$ and $t_1' = 438.1$ vertically downward (constant entropy) to the line $p_2 = 1$ and read 0.826 at the intersection of this line with the constant quality line (interpolated in this case).

Final volume:
$$v_2 = x_2u_2 + \sigma_2$$
$$= 0.826 \times 333 + 0.016$$
$$= 275 \text{ cu. ft.}$$

(This quantity may be taken directly from the total heat pressure diagram.)

External work: $W = \frac{1}{A}\left[(H_1' - AP_1v_1') - (x_2\rho_2 + q_2)\right]$,

$$= 778\left[\left(1243.1 - \frac{144 \times 115}{778} \cdot 4.51\right) - (0.826 \times 972.9 + 69.8)\right]$$
$$= 213{,}938 \text{ ft. lb.}$$

Heat absorbed from the fluid
$$= H_1 - (x_2r_2 + q_2)$$
$$= 1243.1 - (0.826 \times 1034.6 + 69.8) = 318.8 \text{ B.t.u.}$$

Mollier diagram, Fig. 687: Project the intersection of $p_1 = 115$ and $t_1' = 438.1$ upon the Y axis and read $H_1' = 1243$. Similarly the projection of the intersection of $p_2 = 1$ and $x_2 = 0.826$ gives $H_2 = 924.3$, $H_1' - H_2 = 1243 - 924.3 = 318.7$ B.t.u.

393. Polytropic Change of State. — A general law for the expansion of any vapor (wet, dry, or superheated) is

$$pv^n = \text{constant}, \tag{385}$$

$$p_1v_1^n = p_2v_2^n, \tag{386}$$

$$v_2 = v_1\left(\frac{p_1}{p_2}\right)^{\frac{1}{n}}. \tag{387}$$

By giving n special values we are able to obtain the various changes of state for constant volume, constant pressure, isothermal and adiabatic.

The work done by expansion for all values of n, except n = 1, may be expressed

$$W = \int_1^2 P\,dv^n \tag{388}$$

$$= \frac{P_1v_1 - P_2v_2}{n-1} \tag{389}$$

For n = 1,

$$W = \int_1^2 P\,dv \tag{390}$$

$$= P_1v_1 \log_e \frac{v_2}{v_1}. \tag{391}$$

Saturated Steam. — Since with wet or saturated steam there can be no change of pressure without a change of temperature, the value of n will vary with every change of state and for this reason the use of equations (385) and (388) are more troublesome than the preceding thermal analysis. An exception is that of "saturated expansion" in which steam remains saturated throughout change of state. A study of the actual volume occupied by a pound of dry steam at various pressures will show that n has an approximately constant value of 1.0646 or,

$$p_1u_1^{1.0646} = \text{constant}, \tag{392}$$

$u = s - \sigma$. (Except for high pressures the influence of σ is negligible and $u = s$ may be safely assumed.)

This condition of constant saturation during expansion seldom occurs in steam engine practice but equation (392) offers the only simple solution of problems involving work done by such a change of state.

Example 102. — One pound of steam at an initial pressure of 115 lb. per sq. in. absolute expands to a pressure of 2 lb. absolute and maintains a saturated condition throughout expansion. Required the final volume and the work done during expansion.

Solution. — From equations (386) and (392)

$$u_2 = u_1\left(\frac{p_2}{p_1}\right)^{\frac{1}{n}}$$

$$= (3.88 - 0.018)\left(\frac{115}{2}\right)^{\frac{1}{1.0646}}$$

$$= 173.6 \text{ cubic feet.}$$

This value checks with that obtained from steam tables.

Work done $W = \frac{P_1 u_1 - P_2 u_2}{n - 1}$

$$= \frac{144\ (115 \times 3.862 - 2 \times 173.6)}{1.0646 - 1} = 216{,}000.$$

Wet Steam. Actual Expansion. — The values of n for the expansion and compression curves of indicator diagrams from actual engines are subject to a wide variation. A study of several types and sizes of engines by J. Paul Clayton[1] gave values of n varying from 0.7 for wet steam to 1.34 for highly superheated steam. The *average* value of n is, however, not far from 1. That n = 1 for isothermal gas expansion and the average actual steam cylinder expansion is a mere coincidence and does not signify that the expansion in the latter is isothermal. See Conventional Diagram, par. 399.

Example 103. — One pound of saturated steam at an initial pressure of 115 lb. per sq. in. absolute expands so that its volume has been increased 5 times. Required the work done during expansion.[2]

Solution. — $W = P_1 v_1 \log_e \frac{v_2}{v_1}$

$$= 144 \times 115 \times 3.88 \log_e \tfrac{5}{1},$$
$$= 103{,}200 \text{ ft. lb.}$$

Wet Steam. Adiabatic Expansion. — The ease with which problems involving adiabatic expansion of vapor or moderately superheated steam can be solved by exact thermal analysis precludes the use of the more troublesome polytropic expansion law. A number of attempts have been made to derive laws which will give the value of n for adiabatic expansion of saturated or wet steam but their accuracy is limited to a comparatively narrow range of pressures and quality. A rule formulated by H. E. Stone[3] and often used in this connection is:

$$n = 1.059 - 0.000315\, p + (0.0706 + 0.000376\, p)\, x. \qquad (393)$$

Example 104. — One pound of steam expands adiabatically from an initial pressure of 115 lb. per sq. in. and quality 0.9 to a pressure of 1 lb. absolute. Required the final volume and the work done during expansion by exact thermal methods and by the polytropic law using equation (393) for determining the value of n.

Solution. — From steam tables:

$p_1 = 115,\ q_1 = 309,\ \rho_1 = 797.9,\ \theta_1 = 0.4877,\ n_1 = 1.103,\ v_1 = 3.88,$
$p_2 = 1,\ q_2 = 69.8,\ \rho_2 = 972.9,\ \theta_2 = 0.1327,\ n_2 = 1.8427,\ v_2 = 333.$

[1] University of Illinois Bulletin, Vol. 9, No. 26, 1915.

[2] Assuming $n = 1$.

[3] University of Illinois Bulletin, Vol. 9, No. 26, p. 79.

Exact thermal methods:

$$
\begin{aligned}
x_2 &= \frac{x_1 n_1 + \theta_1 - \theta_2}{n_1} \\
&= \frac{0.9 \times 1.103 + 0.4877 - 0.1327}{1.8427} \\
&= 0.731. \\
v_2 &= x_2 u_2 + \sigma_2 \\
&= 0.731\ (333 - 0.016) + 0.016 \\
&= 247.7 \text{ cu. ft.} \\
W &= \frac{1}{A}\left[(x_1 \rho_1 + q_1) - (x_2 \rho_2 + q_2)\right] \\
&= 778\left[(0.9 \times 797.9 + 309) \quad (0.731 \times 972.9 + 69.8)\right] \\
&= 191{,}571 \text{ ft.-lb.}
\end{aligned}
$$

Polytropic law:

$$
\begin{aligned}
\text{n} &= 1.059 - 0.000315 \times 115 + (0.0706 + 0.000376 \times 115)\ 0.9 \\
&= 1.125. \quad \text{(From Equation 393).} \\
v_1 &= x_1 u_1 + \sigma_1 = 0.9 \times (3.88 - 0.016) + 0.016 \\
&= 3.5 \text{ cubic feet.}
\end{aligned}
$$

$$
\begin{aligned}
p_1 v_1^{\text{n}} &= p_2 v_2^{\text{n}}. \\
115 \times 3.5^{1.125} &= 1 \times v_2^{1.125}, \\
v_2 &= 235.6 \text{ cu. ft.}
\end{aligned}
$$

$$
\begin{aligned}
W &= \frac{P_1 v_1 - P_2 v_2}{n - 1} \\
&= \frac{144\ (115 \times 3.5 - 1 \times 235.6)}{1.125 - 1} \\
&= 192{,}268 \text{ ft.-lb.}
\end{aligned}
$$

The value of n which will give the same work during expansion according to the polytropic law as the exact thermal analysis, for the conditions specified in the problem, may be determined as follows:

$$
\begin{aligned}
W &= \frac{P_1 v_1 - P_2 v_2}{\text{n} - 1}, \\
191{,}571 &= \frac{144\ (115 \times 3.5 - 1 \times 247.7)}{\text{n} - 1}, \\
\text{n} &= 1.11.
\end{aligned}
$$

This value of n is an *average* only since the true value varies at different points along the expansion line. This may be shown by plotting the true adiabatic expansion line on logarithmic cross-section paper. See par. 400.

Superheated Steam. Isothermal Expansion. — For steam so highly superheated that it does not approach the wet state at any point during the change of state, n = 1, and the exponential law offers the only simple solution for the work done during expansion. This case has been treated in par. 390.

Superheated Steam. Adiabatic Expansion. — The work done during

adiabatic expansion may be approximated from the polytropic law by making n = 1.3. Goodenough gives the following as more accurate than the simple law pv^n = constant.

$$p\ (v' + 0.088)^{1.31} = \text{constant.} \tag{394}$$

Example 105.—Steam at 60 lb. per sq. in. absolute pressure and initially superheated to 300 deg. fahr. expands to a pressure of 15 lb. absolute. Required the final volume and work done according to the polytropic law.

Solution.—From superheated steam tables for $p_1 = 60$ and superheat of 300 deg. fahr.

$$v_1' = 10.41,$$
$$60\ (10.41 + 0.088)^{1.31} = 15\ (v_2' + 0.088)^{1.31},$$
$$v_2' = 30.2.$$

Thermal analysis gives $v_2' = 30$.

$$W = \frac{P_1\ (v_1' + 0.088) + P_2\ (v_2' + 0.088)}{n - 1}$$
$$= \frac{144\ (60 \times 10.5 + 15 \times 30.1)}{1.31 - 1}$$
$$= 83{,}000 \text{ approx.}$$

Thermal analysis gives $W = 78{,}800$.

CHAPTER XXIII. — SUPPLEMENTARY

ELEMENTARY THERMODYNAMICS OF THE STEAM ENGINE

394. General. — The recent marked improvement in the heat economy of the piston engine is largely due to a better understanding of the thermodynamic principles involved in its operation. Once the engine has been constructed, no amount of attention or mechanical adjustment will appreciably affect the economy since the heat efficiency is primarily a function of the design. It is not the object of this chapter to analyze the various thermodynamic laws underlying the design and operation of the piston engine but rather to show their application to the existing types of steam prime movers. In developing an engine with a view of bettering the performance, a knowledge of the theoretical limitations of the particular type under consideration is necessary. With this limit as a guide, the degree of perfection of the actual mechanism is readily ascertained by comparing test results with those theoretically obtainable. Complete conversion of the heat supplied into useful work is impossible for even the perfect or ideal engine; hence some other standard than the heat supplied is desirable for comparison. There are several ideal cycles which simulate to a certain extent the action of steam in the real engine. The more important of these will be treated in detail.

395. Carnot Cycle. — The Carnot cycle gives the highest possible efficiency for any type of heat and it would seem to be the most desirable cycle for the steam engine; but, as will be shown later, there are practical limitations which more than offset the thermodynamic advantage. Nevertheless a study of this cycle is of importance in showing the absolute degree of perfection which can be realized theoretically.

Notations:

$A = \frac{1}{778}$. p = lb. per sq. in. abs.

P = lb. per sq. ft. abs. x = quality of wet steam.

s = specific volume of dry steam, cu. ft. per lb.

v = specific volume of vapor, cu. ft. per lb.

σ = specific volume of water, cu. ft. per lb.

u = increase in volume during evaporation, cu. ft.

t = deg. fahr. above zero. T = deg. fahr. abs.

c_m = mean specific heat of water.

C = mean specific heat of superheated steam.

H = heat content above 32 deg. fahr., B.t.u. per lb.

λ = total heat of dry steam, B.t.u. per lb.

r = latent heat of vaporization, B.t.u. per lb.

ρ = internal latent heat, B.t.u. per lb.

q = heat of liquid, B.t.u. per lb.
θ = entropy of the liquid.
n = entropy of the vapor.
N = total entropy.
Prime marks indicate superheat.

Subscripts *1*, *2*, *w*, *s* indicate, respectively, initial condition, final condition, wet steam, and superheated steam.

The diagram in Fig. 688 represents the pressure-volume action of an ideal steam engine operating in the Carnot cycle.[1] For simplicity assume the cylinder to be 1 sq. ft. in area, to contain unit weight of water and to have a piston displacement equivalent to 1 lb. of saturated steam at the existing back pressure. At the beginning of the stroke *0*, the non-conducting cylinder contains water at temperature T_1 corresponding to pressure P_1. Heat is added to the liquid until vaporization is complete, the movement of the frictionless piston being such that the pressure and therefore the temperature is constant, that is, expansion from *0* to *1* is *isothermal*. The source of heat is now removed and the piston is forced from *1* to *2* by the expansion of the steam. Since the cylinder is non-conducting and there is no reception or rejection of heat, the expansion from *1* to *2* is *adiabatic*. From *2* to *3* heat is abstracted from the steam at such a rate that the temperature and hence the pressure remains constant, that is, the steam is compressed *isothermally*. At *3* the heat abstraction is terminated and the mixture of vapor and liquid is compressed *adiabatically* to the initial temperature and pressure T_1. The location of point *3* is such that water only at temperature T_1 will be present at the end of compression. This assumption that there is only water at *0* and saturated steam at *1* is not necessary, and any degree of wetness or superheat may be assumed since it in no way affects the efficiency.

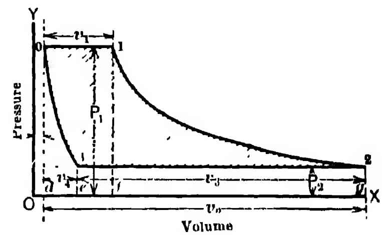


FIG. 688. P-V Diagram for Perfect Engine Operating in the Carnot Cycle.

The net work per cycle is represented by the shaded area *0123*.

$$\text{Area } 0123 = \text{area } 01fd + \text{area } 12gf - \text{area } 32gc - \text{area } d03c \qquad (395)$$

$$\text{Area } 01fd = P_1 v_1 = P_1 (s_1 - \sigma_1) = P_1 u_1. \quad \text{See equation (347).}$$

Since no heat is added during expansion from *1* to *2*, the internal work is equal to the difference in intrinsic energy. See equation (379), hence:

$$\text{Area } 12gf = [(\rho_1 + q_1) - (x_2\rho_2 + q_2)]\frac{1}{A} \qquad (396)$$

[1] This is really a diagram for the complete power plant, involving engine, boiler, condenser, feed pump, etc., and not an indicator card of the cylinder.

$$\text{Area } 32ge = P_2v_3 = P_2v_2 - P_2v_4. \tag{397}$$

But $\qquad v_2 = x_2u_2 + \sigma_2$ (see equation (310))

and $\qquad v_4 = x_3u_2 + \sigma_2.$

Substituting these values in equation (397)

$$\begin{aligned}\text{Area } 32ge &= P_2x_2u_2 - P_2x_3u_2 \\ &= P_2u_2\,(x_2 - x_3).\end{aligned}$$

Since no heat is added during compression from *3* to *0* and there is only liquid at *0* the external work done on the steam is equal to the increase in intrinsic energy, or

$$\text{Area } d03e = [q_1 - (x_3\rho_2 + q_2)]\,\frac{1}{A}.$$

All of these factors with the exception of x_2 and x_3 may be obtained directly from the steam tables. x_2 and x_3 may be calculated from equation (374) or they may be taken directly from the temperature-entropy diagram.

From the above data the PV diagram may be readily plotted to scale. In order to obtain the true contour of the expansion and compression lines, several intermediate points should be calculated and located on the diagram.

The area *0123* when correctly drawn should check with the calculated work. Substituting the values of the different areas in equation (395), we have

$$\begin{aligned}\text{Net work per cycle} &= P_1u_1 + [(\rho_1 + q_1) - (x_2\rho_2 + q_2)]\,\frac{1}{A} - P_2u_2\,(x_2 - x_3) \\ &\qquad - [q_1 - (x_3\rho_2 + q_2)]\,\frac{1}{A} \\ &= P_1u_1 + \frac{\rho_1}{A} - x_2\left(P_2u_2 + \frac{\rho_2}{A}\right) + x_3\left(P_2u_2 + \frac{\rho_2}{A}\right).\end{aligned} \tag{398}$$

Heat absorbed in doing work

$$\begin{aligned}&= AP_1u_1 + \rho_1 - x_2\,(AP_2u_2 + \rho_2) + x_3\,(AP_2u_2 + \rho_2), \\ &= AP_1u_1 + \rho_1 - (x_2 - x_3)\,(AP_2u_2 + \rho_2).\end{aligned} \tag{399}$$

From equation (325) $AP_1u_1 + \rho_1 = r_1$ and $AP_2u_2 + \rho_2 = r_2$.
Therefore heat absorbed

$$= r_1 - r_2\,(x_2 - x_3). \tag{400}$$

The water rate or steam consumption per hp-hr. of the ideal engine working in this cycle is

$$W = \frac{\text{Heat equivalent of 1 hp-hr.}}{\text{Heat absorbed per lb. of fluid}} \tag{401}$$

$$= \frac{2547}{r_1 - r_2\,(x_2 - x_3)}. \tag{402}$$

Efficiency:

$$E = \frac{\text{Heat absorbed}}{\text{Heat supplied}} \tag{403}$$

$$= \frac{r_1 - r_2\,(x_2 - x_3)}{r_1}. \tag{404}$$

But $$r_2\,(x_2 - x_3) = \frac{T_2}{T_1}r_1, \text{ see equation (368).} \tag{405}$$

Therefore $$E = \frac{r_1 - \frac{T_2}{T_1}r_1}{r_1} = \frac{T_1 - T_2}{T_1}, \tag{406}$$

which is independent of the nature of the working substance and dependent only on the range of temperature.

The shaded area *0123*, Fig. 689, represents the *P-V* relationship of Fig. 688 plotted in the temperature-entropy diagram in which ordinates are absolute temperatures and abscissas increase of entropy. This diagram is useful in visualizing the thermal changes per stroke or cycle. Line *ww* represents the increase of entropy of the liquid above 32 deg. fahr. and *ss* the increase of entropy of the vapor. Both of these lines are readily constructed by plotting several values of θ and N as abscissas for corresponding values of T as ordinates. These quantities may be taken directly from steam tables. *0-1* therefore represents the isothermal expansion of the fluid from water at temperature T_1 to dry steam at the same temperature. Since the entropy is constant for adiabatic expansion, *1-2* represents the expansion of the saturated fluid from temperature T_1 to temperature T_2. Similarly *2-3* represents isothermal compression at temperature T_2 and *3-0* adiabatic compression from temperature T_2 to the initial condition. If the various lines are drawn to scale

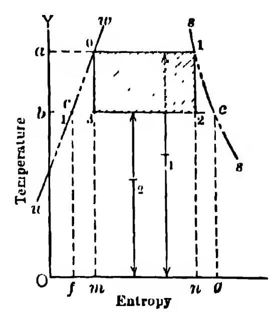


FIG. 689. Temperature-entropy Diagram; Perfect Engine Carnot Cycle.

Heat supplied above 32 deg. fahr. = area *m01n*.

$$\text{Area } m01n = 0\text{–}1 \times T_1 = n_1 T_1 = r_1.$$

Heat rejected above 32 deg. fahr. = area *m32n*.

$$\text{Area } m32n = 3\text{–}2 \times T_2 = n_1 T_2.$$

Heat absorbed = area *0123* = area *m01n* − area *m32n*

$= r_1 - n_1 T_2.$

$= r_1 - r_2 (x_2 - x_3).$

Quality at end of expansion $x_2 = \frac{c2}{ce} = \frac{a0 + 01 - bc}{ce} = \frac{n_1 + \theta_1 - \theta_2}{}.$

Quality at beginning of compression $x_3 = \frac{c3}{ce} = \frac{a0 - bc}{ce} = \frac{\theta_1 - \theta_2}{n_2}.$

For any degree of wetness at the beginning and end of isothermal expansion, the point *0* will lie to the right of the intersection of *ww* and T_1, and the point *1* will lie to the left of the intersection of *ss* and T_1. The figure *0123*, however, will always be a rectangle.

If isothermal application of heat is continued during admission until the fluid is superheated, the point *1* will still lie on the line *a01* but to the right of the vapor line *ss*. In order to maintain a constant temperature of T_1 in the superheated zone, the pressure must be lowered according to the law expressed by equation (308). Since superheat is supplied in practice with gradually increasing temperature and not isothermally, the Carnot cycle is not a satisfactory standard for comparing engines using superheated steam and hence this case will not be considered.

Example 106. — Determine the heat absorbed, water rate and efficiency of a perfect engine working in the Carnot cycle if the cylinder contains only water at the beginning of the cycle and saturated steam at cut off. Initial pressure 215 lb. per sq. in. absolute; back pressure, 2 lb. absolute. Assume one pound of fluid per cycle.

Solution. — From steam tables:

$p_1 = 215,\ t_1 = 388,\ s_1 = 2.138,\ q_1 = 361.4,\ r_1 = 837.9,\ \rho_1 = 754,$
$\theta_1 = 0.5513,\ n_1 = 0.9885,\ \sigma_1 = 0.0185,\ N_1 = 1.5398,$
$p_2 = 2,\ t_2 = 126.15,\ s_2 = 173.5,\ q_2 = 94,\ r_2 = 1021,\ \rho_2 = 956.7,$
$\theta_2 = 0.1749,\ n_2 = 1.7431,\ \sigma_2 = 0.0162.$

Qualities:

$x_0 = \text{zero}. \qquad x_1 = \text{unity}.$

$$x_2 = \frac{N_1 - \theta_2}{n_2} = \frac{1.5398 - 0.1749}{1.7431} = 0.7833. \quad \text{(See equation (374).)}$$

$$x_3 = \frac{\theta_1 - \theta_2}{n_2} = \frac{0.5513 - 0.1749}{1.7431} = 0.216.$$

Specific volumes:

$v_0 = \sigma_1 = 0.0185.$
$v_1 = s_1 - \sigma_1 = 2.138 - 0.0185 = 2.12.$
$v_2 = x_2 u_2 + \sigma_2 = 0.7833 \times 173.5 = 135.9.$ (See note, equation (310).)
$v_3 = v_2 - v_4 = 135.9 - 37.53 = 98.37.$
$v_4 = x_3 u_2 + \sigma_2 = 0.216 \times 173.5 = 37.53.$ (See note, equation (310).)

Work:

$$\text{Admission: } P_1v_1 = 144 \times 215 \times 2.12$$
$$= 65{,}635 \text{ ft-lb.}$$

$$\text{Expansion} = \frac{1}{A}\left[(\rho_1 + q_1) - (x_2\rho_2 + q_2)\right]$$
$$= 778\left[(754 + 361.4) - (0.7833 \times 956.7 + 94)\right]$$
$$= 211\,616 \text{ ft-lb.}$$

$$\text{Exhaust: } P_2v_3 = 144 \times 2 \times 98.37$$
$$= 28{,}350 \text{ ft-lb.}$$

$$\text{Compression} = \frac{1}{A}\left[q_1 - (x_3\rho_2 + q_2)\right]$$
$$= 778\left[361.4 - (0.216 \times 956.7 + 94)\right],$$
$$= 47{,}302 \text{ ft-lb.}$$

$$\text{Net work} = (65{,}635 + 211{,}616) - (28{,}350 + 47{,}302),$$
$$= 201{,}599 \text{ ft-lb.}$$

Heat:

Equivalent of work done $= 201{,}599 \div 778 = 259.1$ B.t.u.
Supplied $= r_1 = 837.8$ B.t.u.

$$\text{Efficiency: } \quad E_r = \frac{259.1}{837.9} = 0.309 = 30.9 \text{ per cent.}$$

$$\text{Water rate: } W_r = \frac{2546}{259.1} = 9.83 \text{ lb. per hp-hr.}$$

Temperature-Entropy Diagram

$$\text{Heat equivalent of work done} = n_1\,(T_1 - T_2) = n_1\,(t_1 - t_2)$$
$$= 0.9885\,(388 - 126.15)$$
$$= 259.0 \text{ B.t.u.}$$

$$\text{Efficiency} = \frac{T_1 - T_2}{T_1} = \frac{261.85}{848} = 0.309 = 30.9 \text{ per cent.}$$

While it is conceivable to build an engine which will simulate the true Carnot cycle it would be practically impossible to do so without introducing evils which would more than counterbalance the thermodynamic advantage. The compression in the actual engine must not be confused with the adiabatic compression of the Carnot cycle, since the cushion steam involved in the operation of the former is but a fraction of the total fed to the cylinder and has but little influence on the thermodynamic action of the engine.

A modification of the Carnot cycle, known as the *regenerative steam-engine cycle* and having the same efficiency as the Carnot cycle, has been simulated by a special type of Nordberg pumping engine. The engine is quadruple-expansion with four cylinders, three receivers and five feed-water heaters in series *a*, *b*, *c*, *d*, and *e*. The feedwater is taken from the hotwell and passed in succession through the various heaters: *a* receives

its heat from the exhaust steam on its passage to the condenser; b receives its heat from the low-pressure cylinder jacket; and c, d, and e, respectively, from the third, second, and first receivers. Referring to Fig. 690, if $1-c'$ is drawn parallel to the water line ww the area $01c'c$ will equal the area of the Carnot cycle 0123. The Nordberg engine approximates this cycle as indicated by the broken lines. The expansion in the first stage corresponds to $1-a_1$, that in the second to a_1-a_2, and so on for each of the other stages. Heat represented by the area below a_1-a_1' is abstracted from the first stage and is used to raise the condition of the water from b_2' to b_1; heat corresponding to the area below a_2a_2' is withdrawn from the second stage and is used to raise the condition of the water from b_3 to b_2; and so on for each stage. Thus heat is abstracted by steps from the expanding steam and is used for progressively heating the feedwater. The greater the number of steps the nearer will the actual cycle approach the ideal. The famous Nordberg compressor[1] attained 73.7 per cent of the efficiency of the Carnot cycle for the same temperature limits, an exceptional performance for this period.

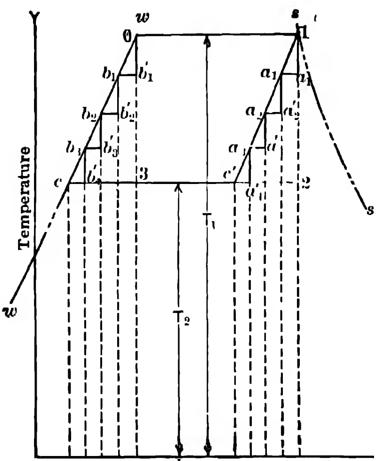


FIG. 690. Regenerative Steam Engine Cycle.

396. Rankine Cycle. Complete Expansion.[2] — This cycle has been adopted by the American Society of Mechanical Engineers and the British Institution of Civil Engineers as the standard for comparing the performance of all steam prime movers. It is of value not only in comparing the performances of steam engines with each other but also in comparing engines with turbines. In an engine working according to the Rankine cycle, steam is admitted at constant pressure, expanded adiabatically to

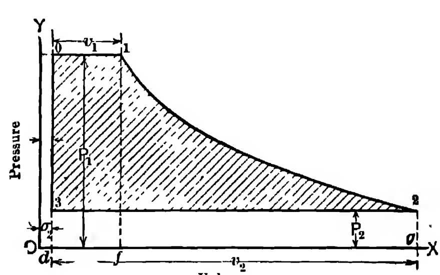


FIG. 691. Indicator Card for Perfect Engine Working in the Rankine Cycle with Complete Expansion.

[1] Eng. News, May 4, 1899, p. 280.

[2] This is often called the Clausius cycle since it was published simultaneously but independently by both Clausius and Rankine.

the back pressure, and exhausted at that pressure. The engine has no clearance and there are no heat losses from friction, imperfect expansion, or otherwise, all the energy taken from the steam being converted into work. The diagram *0123*, Fig. 691, represents the familiar indicator card or pressure-volume diagram of the working fluid operating in this cycle. *0–1* represents the admission of steam from the boilers at constant pressure P_1; *1–2* is an adiabatic expansion to exhaust pressure P_2; *2–3* exhaust at constant pressure P_2; and *3–0* a practically constant volume pressure rise.

For all conditions of steam:

$$
\begin{aligned}
\text{Work done during admission} &= \text{area } 01fd\\
\text{Work done during expansion} &= \text{area } 12gf\\
\text{Work done during exhaust} &= \text{area } 32gd\\
\text{Net work} &= \text{area } 01fd + \text{area } 12gf - \text{area } 32gd\\
&= \text{area } 0123
\end{aligned}
$$

Per pound of wet or saturated steam:

$$
\begin{aligned}
\text{Work done during admission} &= P_1\,(x_1u_1 + \sigma_1)\ \text{ft-lb.}\\
\text{Work done during expansion} &= \frac{1}{A}\left[(x_1\rho_1 + q_1) - (x_2\rho_2 + q_2)\right]\ \text{ft-lb.}\\
\text{Work done during exhaust} &= P_2\,(x_2u_2 + \sigma_2)\ \text{ft-lb.}\\
\text{Net work} &= P_1\,(x_1u_1 + \sigma_1) + \frac{1}{A}\left[(x_1\rho_1 + q_1)\right.\\
&\quad \left. - (x_2\rho_2 + q_2)\right] - P_2\,(x_2u_2 + \sigma_2)\ \text{ft-lb.} \qquad (407)\\
&= x_1r_1 + q_1 - (x_2r_2 + q_2)^*\ \text{B.t.u.} \qquad (408)\\
&= H_1 - H_2\ \text{B.t.u.} \qquad (409)
\end{aligned}
$$

Per pound of steam superheated at admission but wet or saturated at end of expansion:

$$
\begin{aligned}
\text{Work done during admission} &= P_1v_1'\ \text{ft-lb.}\\
\text{Work done during expansion} &= \left(\frac{1}{A}\,H_1' - P_1v_1'\right) - \frac{1}{A}\,(x_2\rho_2 + q_2)\ \text{ft-lb.}\\
\text{Work done during exhaust} &= P_2\,(x_2u_2 + \sigma_2)\ \text{ft-lb.}\\
\text{Net work} &= P_1v_1' + \left[\left(\frac{1}{A}\,H_1' - P_1v_1'\right) - (x_2\rho_2 + q_2)\right]\\
&\quad - P_2\,(x_2u_2 + \sigma_2)\ \text{ft-lb.}\\
&= H_1' - (x_2\rho_2 + q_2) - A\,P_2(x_2u_2 + \sigma_2)\ \text{B.t.u.}\\
&= H_1' - (x_2r_2 + q_2)^*\ \text{B.t.u.} \qquad (410)\\
&= H_1' - H_2\ \text{B.t.u.} \qquad (411)
\end{aligned}
$$

* The quantities $P_1\sigma_1$ and $P_2\sigma_2$ are negligible and have been omitted in this equation.

Per pound of steam superheated throughout admission and expansion:

Work done during admission $= P_1v_1'$ ft-lb.

Work done during expansion $= \frac{1}{A} H_1' - P_1v_1' - \left(\frac{1}{A} H_2' - P_2v_2\right)$ft-lb.

Work done during exhaust $= P_2v_2'$ ft-lb.

$$\text{Net work} = P_1v_1' + \frac{1}{A}(H_1' - H_2') - P_1v_1' + P_2v_2' - P_2v_2' \text{ ft-lb.} \quad (412)$$

$$= H_1' - H_2' \text{ B.t.u.} \quad (413)$$

Calling H_i and H_n the initial and final heat content for all conditions of steam, a general expression for the heat converted into work H_w is

$$H_w = H_i - H_n. \quad (414)$$

Heat supplied H_t above exhaust temperature t is

$$H_t = H_i - q_n. \quad (415)$$

$$\text{Efficiency } E_r = \frac{H_i - H_n}{H_i - q_n} \quad (416)$$

Steam consumption or water rate, lb. per hp-hr., is

$$W_r = \frac{2547}{H_i - H_n} = \frac{2547}{E_r(H_i - q_n)}. \quad (417)$$

The temperature-entropy diagrams for the conditions discussed above are shown in Figs. 692 to 694. For saturated or wet steam it will be

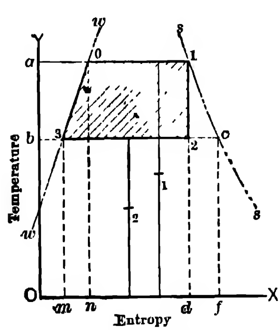


FIG. 692. Temperature-entropy Diagram; Perfect Engine, Rankine Cycle with Complete Expansion. Steam Dry at Cut-off.

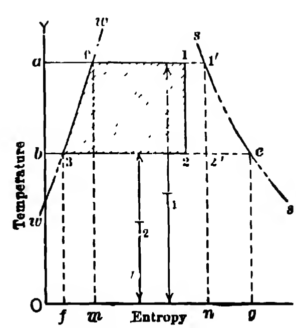


FIG. 693. Temperature-entropy Diagram; Perfect Engine, Rankine Cycle for Wet Steam at Cut-off.

noted that the admission line is an isothermal since a constant-pressure expansion for saturated steam is also a constant-temperature one. For superheated steam, however, the temperature increases with the degree

of superheat, the pressure remaining constant, and the relation between pressure and volume varies according to the law expressed in equation (308), that is, the location of point 1′, Fig. 694, is fixed by determining the entropy corresponding to pressure P_1 and temperature T_1'. This may be calculated from equation (343) or it may be taken directly from superheated steam tables.

A study of equation (416) in connection with the Mollier diagram will show that

(1) The Rankine cycle when using superheated steam has a lower theoretical efficiency than that of the same cycle with saturated vapor having the same maximum temperature.

(2) The theoretical efficiency increases but slightly with the increase in superheat, the maximum pressure remaining constant; see Table 60.

(3) The theoretical efficiency increases rapidly with the increase in pressure range; see Table 56.

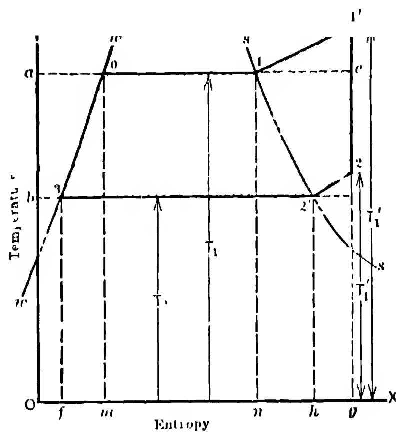


FIG. 694 Temperature-entropy Diagram; Perfect Engine, Rankine Cycle for Steam Superheated throughout Expansion.

The behavior of the actual engine under these conditions is discussed in paragraphs 183 and 186.

A comparison of the Carnot and Rankine cycles shows a lower efficiency for the latter for the same operating conditions, as would be expected. The water rate for the Carnot cycle, however, is higher. This apparent anomaly is due to the fact that the heat supplied per pound of fluid is much larger in the Rankine than in the Carnot. Thus less weight of steam is used per hp-hr., but each pound receives more heat and this is used less efficiently.

Example 107. — A perfect engine operating in the Rankine cycle with complete expansion takes steam at 115 lb. per sq. in. absolute pressure, quality 98, and exhausts against a back pressure of 1 lb. absolute. Required the condition of the steam at end of expansion, the work done, efficiency, and water rate.

Solution. — From steam tables:

$p_1 = 115$, $t_1 = 338.1$, $r_1 = 879.8$, $q_1 = 309$, $H_1 = 1188.8$, $\theta_1 = 0.4877$, $n_1 = 1.103$,

$p_2 = 1$, $t_2 = 101.8$, $r_2 = 1034.6$, $q_2 = 69.8$, $\theta_2 = 0.1327$, $n_2 = 1.8427$,

$$x_2 = \frac{x_1 n_1 + \theta_1 - \theta_2}{n_2} \quad \text{(See equation (374).)}$$

$$= \frac{0.98 \times 1.103 + 0.4877 - 0.1327}{1.8427}$$

$$= 0.779.$$

Heat converted into work

$$= x_1 r_1 + q_1 - (x_2 r_2 + q_2)$$

$$= 0.98 \times 879.8 + 309 - (0.779 \times 1034.6 + 69.8)$$

$$= 1171.2 - 875.7 = 295.5 \text{ B.t.u. per lb.}$$

$$\text{Efficiency} = \frac{H_i - H_2}{H_i - q_2}$$

$$= \frac{295.5}{1171.2 - 69.8} = 0.268 = 26.8 \text{ per cent.}$$

$$\text{Water rate} = \frac{2546}{H_i - H_2}$$

$$= \frac{2547}{295.5} = 8.62 \text{ lb. per hp-hr.}$$

The initial and final heat content may be taken directly from the Mollier diagram; as a matter of fact it is customary in practice to use the diagram except when extreme accuracy is necessary or when the given conditions are beyond the range of the charts.

397. Rankine Cycle with Incomplete Expansion. — If expansion after cut-off is not carried far enough to reduce the pressure to that of the back-pressure line as shown in Fig. 695, the Rankine cycle more nearly simulates the cycle of the actual engine. This cutting the "toe" off the diagram decreases the efficiency, but permits of the use of a smaller cylinder. A comparison of the diagram in Fig. 691 with that in Fig. 695 will show that the area *012'3'* is of the same outline as area *0123*; consequently the work done would be that corresponding to complete expansion to pressure P_c plus that represented by area *3'2'23*.

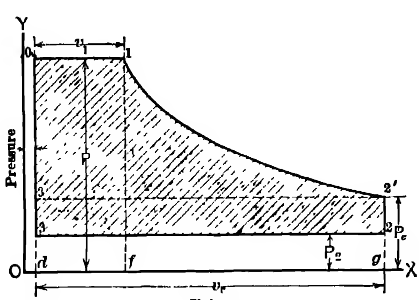


FIG. 695. Indicator Card for Perfect Engine Working in the Rankine Cycle with Incomplete Expansion.

If H_c represents the heat content corresponding to complete expansion to pressure P_c the heat equivalent of the work done (area *012'3'*) is $H_i - H_c$ B.t.u. per lb.

Work corresponding to area *3'2'23* = $(P_c - P_2)\, v_c$ ft-lb. per lb.

Hence, heat converted into work $= H_i - H_c + A\,(P_c - P_2)\,v_c$
Heat supplied is the same as for complete expansion $= H_i - q_2$.

$$\text{Therefore efficiency } E_r' = \frac{H_i - H_c + A\,(P_c - P_2)\,v_c}{H_i - q_2}. \tag{418}$$

$$\text{Water rate } W = \frac{2547}{H_i - H_c + A\,(P_c - P_2)\,v_c} \tag{419}$$

For wet steam, $v_c = x_c u_c + \sigma_c = x_c s_c$ (for all practical purposes).
For dry steam, $v_c = s_c - \sigma_c$.
For superheated steam, $v_c = v_c' - \sigma_c$.

The temperature-entropy diagram differs from that for complete expansion in the curtailment of lines *1-3'* and *3' 3* by constant-volume pressure drop *2'-2*, Fig. 696.

Example 108. — Same data and requirements as in preceding example except that release occurs at a pressure of 4 lb. absolute.

Solution. — From steam tables: p_1 and p_2 as in preceding example,

$$p_c = 4,\ r_c = 1005.7,\ q_c = 120.9,\ \theta_c = 0.2198,$$
$$n_c = 1.6416,\ s_2 = 90.5,$$

$$x_c = \frac{x_1 n_1 + \theta_1 - \theta_c}{n_c} = \frac{0.98 \times 1.103 + 0.4877 - 0.2198}{1.6416} = 0.822.$$

$$v_c = x_c s_2 = 0.822 \times 90.5 = 74.4.$$

$$H_c = x_c r_c + q_c = 0.822 \times 1005.7 + 120.9 = 947.6.$$

$H_1 = 1171.2$ (same as in preceding example).

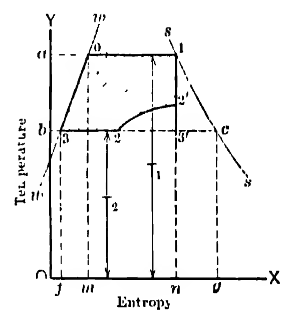


FIG. 696. Temperature-entropy Diagram; Perfect Engine, Rankine Cycle with Incomplete Expansion. Steam Dry at Cut-off.

$$\text{Efficiency} = \frac{H_1 - H_c + A\,(P_c - P_2)\,v_c}{H_1 - q_2}$$
$$= \frac{1171.2 - 947.6 + \frac{144}{778}\,(4 - 1)\,74.4}{1171.2 - 69.8}$$
$$= \frac{1171.2 - 947.6 + 41}{1171.2 - 69.8} = \frac{264.6}{1101.4}$$
$$= 0.24 = 24 \text{ per cent.}$$

$$\text{Water rate} = \frac{2547}{264.6} = 9.62 \text{ lb. per hp-hr.}$$

398. Rankine Cycle with Rectangular PV-Diagram. — This cycle is the least efficient of all vapor cycles in practical use but represents the

action of the fluid in direct-acting steam pumps, direct-acting air compressors and engines taking steam full stroke. It may be looked upon as a limiting case of the Rankine cycle. From Fig. 697 it is apparent that

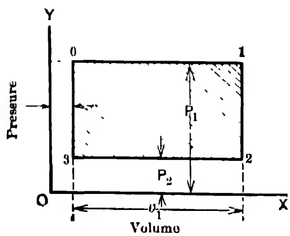


FIG. 697.

$$\text{Work done} = A\ (P_1 - P_2)\ v \text{ B.t.u.} \qquad (420)$$

For wet steam, $v = x_1u_1 + \sigma_1 = x_1s_1$ (for most purposes).

For dry steam, $v = s_1 - \sigma_1$.

For superheated steam, $v = v_1' - \sigma_1$.

Heat received is the same as that in the Rankine cycle

$$= H_i - q_n.$$

$$\text{Efficiency} = \frac{A\ (P_1 - P_2)\ v}{H_i - q_n} \qquad (421)$$

$$\text{Water rate} = \frac{2547}{A\ (P_1 - P_2)\ v} \qquad (422)$$

Example 109. — A perfect direct-acting steam pump operating in the rectangular PV cycle takes steam at initial pressure 115 lb. per sq. in. absolute, quality 98 per cent and exhaust against a back pressure of 15 lb. absolute. Required the work done per lb. of fluid, efficiency and the water rate.

Solution. — From steam tables:

$$p_1 = 115,\ s_1 = 3.88,\ H_1 = 1188.8,$$

$$p_2 = 15,\ q_n = q_2 = 181.0.$$

$$\begin{aligned}\text{Heat converted into work} &= A\ (P_1 - P_2)\ x_1s_1 \\ &= \tfrac{144}{778}\ (115 - 15)\ 0.98 \times 3.88 \\ &= 70.4 \text{ B.t.u.}\end{aligned}$$

$$\text{Efficiency} = \frac{70.4}{1188.8 - 180} = 0.07 \text{ approx.} = 7 \text{ per cent.}$$

$$\text{Water rate} = \frac{2547}{70.4} = 36 \text{ lb. per hp-hr.}$$

399. Conventional Diagram. — In designing an engine it is customary to assume as a basis of reference an ideal cycle which considers only the kinetic action of the steam in the cylinder. This permits of analysis without the use of steam tables. The expansion is assumed to be hyperbolic because the equilateral hyperbola is readily constructed and because expansion in the actual engine conforms approximately to the law $Pv^n = C$ (see paragraph 393). According to the 1915 A.S.M.E. Code the

ideal engine is assumed to have no clearance and no losses through wire-drawing during admission or release. The initial pressure is that of the boiler and the back pressure that of the atmosphere for a non-condensing engine, and of the condenser for a condensing engine. Such a diagram for a simple non-condensing engine is illustrated in Fig. 698. *0–1* repre-

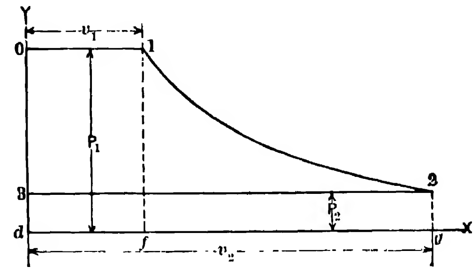


FIG. 698.

sents admission at constant pressure P_1, *1–2* represents hyperbolic expansion from cut-off *1* to release at *2*, and *2–3* represents exhaust at atmospheric pressure P_2.

The work done is represented by the

$$
\begin{aligned}
\text{area } 0123 &= \text{area } 01fd + \text{area } 12gf - \text{area } 32gd,\\
\text{area } 01fd &= P_1v_1,\\
\text{area } 12gf &= P_1v_1 \log_e \frac{v_2}{v_1} \text{ (see paragraph 393)},\\
\text{area } 32gd &= P_2v_2.
\end{aligned}
$$

Therefore net work done

$$W = P_1v_1\left(1 + \log_e \frac{v^2}{v^1}\right) - P_2v_2, \tag{423}$$

letting

$$\frac{v_2}{v_1} = r = \text{ratio of expansion},$$

$$W = P_1v_1 \,(1 + \log_e r) - P_2v_2. \tag{424}$$

$$\text{Mean effective pressure } P_m = \frac{\text{area } 0123}{v_2}$$

$$= \frac{P_1}{r}(1 + \log_e r) - P_2. \tag{425}$$

As the m.e.p. is generally used in pounds per square inch, dividing both members of the equation by 144 gives

$$p_m = \frac{p_1}{r}(1 + \log_e r) - p_2. \tag{426}$$

$$\text{Theoretical maximum horsepower} = \frac{p_m lan}{33{,}000}, \tag{427}$$

in which

l = length of stroke, ft.,
a = area of piston, sq. in.,
n = number of working strokes.

The ratio of the m.e.p. of the actual engine to that of the ideal diagram as determined above is called the diagram factor. This factor is determined by experiment and ranges as follows (Heat Power Engineering, Hirshfeld and Barnard, 1915, p. 325):

Simple slide-valve engine	55 to 90 per cent
Simple Corliss engine	85 to 90 " "
Compound slide-valve engine	55 to 80 " "
Compound Corliss engine	75 to 85 " "
Triple-expansion engine	55 to 70 " "

The probable mean effective pressure for the engine under consideration is

$$\text{M.e.p.} = p_m \times \text{diagram factor}. \tag{428}$$

Example 110. — Determine the probable horsepower of a 12 inch × 12 inch simple engine, 250 r.p.m., initial pressure 120 lb. per sq. in. absolute, cut off ¼ stroke, diagram factor 0.75.

Solution. —

$$\begin{aligned} \text{Theoretical m.e.p.} &= \tfrac{120}{4}\,(1 + \log_e 4) - 15, \\ &= 56.53. \\ \text{Probable actual m.e.p.} &= 56.53 \times 0.75 = 42.4. \\ \text{Probable i.hp.} &= \frac{42.4 \times 1 \times 113 \times 500}{33{,}000} \\ &= 72.4 \end{aligned}$$

400. Logarithmic Diagram. — It is a well-known fact that the equation of the polytropic curve $Pv^n = C$ becomes a straight line when plotted on logarithmic cross-section paper and the slope of the line is the value of n. Conversely, when the expansion or compression curve of an indicator becomes a straight line in the logarithmic diagram it shows that the change of state is in accordance with the law $Pv^n = C$. The logarithmic diagram derived from the indicator card is useful in analyzing cylinder performance and gives valuable information which cannot be readily obtained otherwise. Thus it has been demonstrated[1] that the logarithmic diagram is of great assistance in

[1] A New Analysis of the Cylinder Performance of Reciprocating Engines. J. Paul Clayton, Univ. of Ill. Bull. No. 26, Vol. 9, May 6, 1912.

(1) Approximating clearance volume.
(2) Locating the stroke positions of cyclic events.
(3) Detecting leakage.
(4) Approximating steam consumption.

Construction of he Logarithmic Diagram. — If the clearance volume is given, the construction of the diagram is very simple. Draw the clearance line OY and the absolute pressure line OX on the indicator diagram as illustrated in Fig. 699. Locate points *1*, *2*, *3*, etc. on the expansion line and tabulate the corresponding absolute pressures and volumes. For example, the pressure corresponding to point *1* is P_1 and its value is the length of the line P_1 multiplied by the scale of the indicator spring.

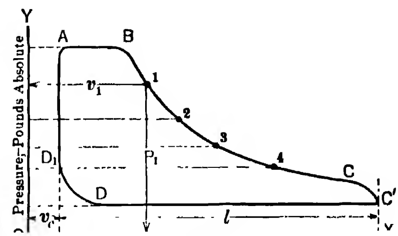


FIG. 699.

Similarly the volume corresponding to point *1* is v_1 and its value is the length of the line v_1 multiplied by constant m (= piston displacement per stroke in cu. ft. divided by the length of the card l measured in inches). Transfer these points to logarithmic cross-section paper as illustrated in Fig. 700, using absolute pressures in lb. per sq. in. as ordinates and cu. ft. as abscissas. Repeat the operation for the compression curve and draw a smooth line through the various points. The ratio $\frac{ab}{bc}$ (measured in inches) will be the value of n for the expansion line and $\frac{de}{ef} = n$ for compression.

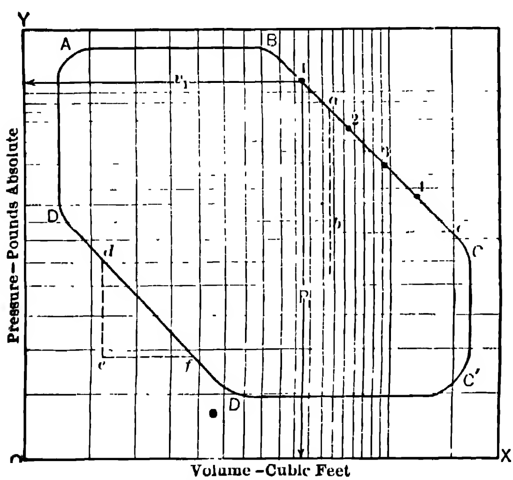


FIG. 700. Indicator Card — Logarithmic Diagram.

Approximating Clearance Volume. — If expansion and compression vary substantially according to the law $Pv^n = C$ the clearance volume may be approximated by trial and error. All that is necessary is to assume different values of clearance and to plot the logarithmic diagram for each assumed value until the expansion or compression curve is a straight line.

Locating the Stroke Position of Cyclic Events. — Except with a few types of four-valve engines it is difficult and oftentimes impossible to locate the points of cut-off, release, and compression from the indicator diagram. If there is no leakage the true points may be located on the logarithmic diagram by noting when the expansion and compression curves become straight; see Fig. 700.

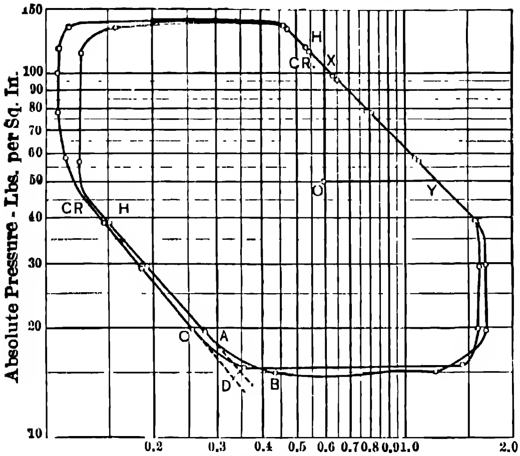


FIG. 701. Logarithmic Diagrams - 12 by 24 Corliss.

Detecting Leakage. — The law $Pv^n = C$ is applicable only to cases where the weight of steam remains practically constant during change of state. When the weight changes materially as by leakage, the resulting expansion and compression lines on the logarithmic diagram depart from straight lines. This is clearly shown in Fig. 703.

Approximating Steam Consumption. — According to Clayton (1) there is a definite relation existing between x_c (quality at cut-off) and n in any one cylinder which is practically independent of cut-off position. (2) This relation is practically independent of cylinder size and of engine speed; it is therefore applicable to other cylinders of the same type. (3) By means of the experimentally determined relations of x_c and n, the value of x_c may be approximated

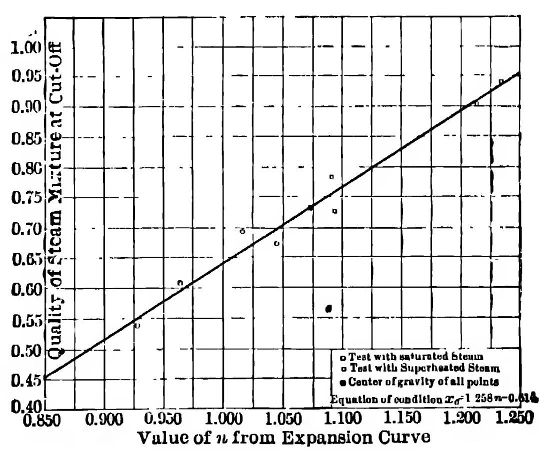


FIG. 702. Relation of Quality and the Value of n.

from the average value of n obtained from the expansion curves of one set of indicator diagrams taken simultaneously; therefore the actual weight of steam present in one revolution may be approximated.

(4) The actual steam consumption may be obtained by this method from the indicator diagram to within an average of 4 per cent of test measurements. These statements apply strictly to non-jacketed steam cylinders in good condition, exhausting at or near atmospheric pressure. In applying this method it is only necessary to determine n as previously outlined and find from the curve in Fig. 702 the corresponding value of x_c. When the quality of steam at cut-off is known, the weight of fluid per stroke can be readily calculated. It will be noted that the curve in Fig. 702 is only an average approximation and that there is a considerable range in the values of x_c for a given value of n. By separating the points into groups of similar pressures and speeds, several lines coördinating n and x_c may be obtained and a greater accuracy is possible. For a complete discussion of this important subject consult Clayton's paper.

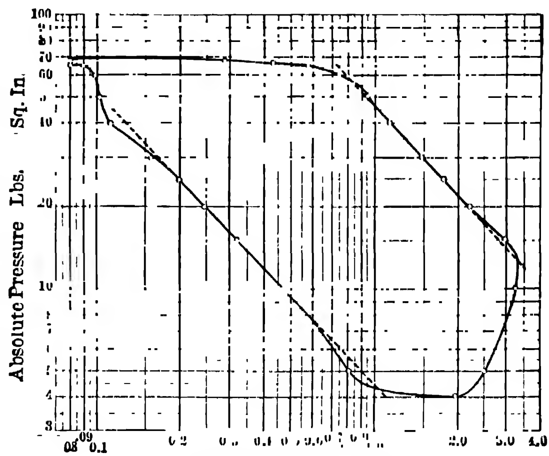


FIG. 703 Diagram from a 14-in by 35-in. Corliss Engine, Showing Leakage at Beginning and End of Expansion and Compression.

401. Temperature-Entropy Diagram. — If the actual indicator card is transferred to the temperature-entropy chart the various heat exchanges during expansion and compression may be seen at a glance. The area represented by the actual diagram, however, does not give the heat utilized in doing work, since the weight of steam is not constant throughout the cycle. From cut-off to release the weight is constant if there is no leakage, as is the case from beginning of compression to admission, but the weights involved in each case are not the same. Therefore, only the expansion line shows the true behavior of all the steam used per cycle and the rest of the diagram is more or less conventional. The transfer of the pressure-volume

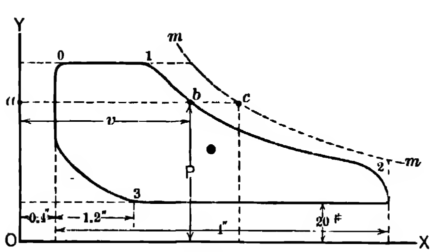


FIG. 704.

to the temperature-entropy diagram is best illustrated by a specific example.

Example 111. — Curve *0123*, Fig. 704, is an average indicator card taken from a 12 × 12 engine running at 300 r.p.m.; clearance volume 10 per cent; steam consumption by tests 2700 lb. per hr. Transfer the indicator card to temperature-entropy chart.

Solution. — Locate the zero clearance line OY and zero pressure lines OX, and measure the diagram as indicated. The piston displacement per stroke $= \dfrac{3.14 \times 12^2}{144 \times 4} = 0.785$ cu. ft.

Compression volume $= 0.785\left(\dfrac{1.2 + 0.4}{4}\right) = 0.314$ cu. ft.

Weight of "cushion steam" on the assumption that the steam is dry at the beginning of compression

$$= 0.314 \times 0.0498 = 0.0156 \text{ lb.}$$

(0.0498 = wt. of 1 cu. ft. of steam at 20 lb. abs. pressure.)
Weight of steam used per stroke or "cylinder feed"

$$= \frac{2700}{600 \times 60} = 0.075 \text{ lb.}$$

Total weight of steam expanding = 0.075 + 0.0156 = 0.09 lb.
Lay off saturation line *mm*. This line represents the volume of 0.09 lb. of saturated steam for the various pressures within the range of the diagram.

Draw several pressure lines such as *abc* and tabulate the ratio $\dfrac{ab}{ac}$. This ratio gives the quality of the steam at point b in the expansion curve $\left(\dfrac{ab}{ac} = \dfrac{xs}{s} = x\right)$. Note that $\dfrac{ab}{ac}$ represents quality only during expansion after cut-off and that it is simply a ratio for other parts of the cycle.

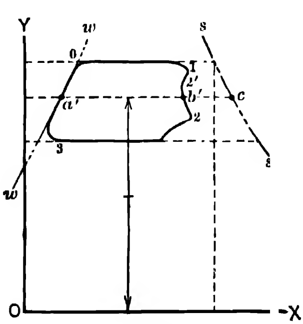


Fig. 705.

Tabulate also the absolute temperature corresponding to the pressure under consideration. Next construct the water and saturation curves *ww* and *ss*, respectively, as illustrated in Fig. 705. This may be done conveniently by using absolute temperatures and entropies of water and vapor given in steam tables, the entropies being multiplied by 0.09. Locate point b' on the corresponding temperature line in such a position that rates $\dfrac{a'b'}{a'c} = \dfrac{ab}{ac}$ obtained from the indicator card. The locus of the point b' will be the desired diagram. The thermal action during actual expansion is apparent from the diagram; thus it will be seen by inspection that the steam is wet at cut-off, that condensation takes place from *1* to *2'*

more rapidly than if expansion were adiabatic, and that reëvaporation takes place from b' to 2.

The foregoing analysis applies only to saturated or wet steam. In case of superheat the actual expansion will lie beyond the saturation curve as illustrated in Fig. 706 and the ratio $\frac{ab}{ac} = \frac{v'}{s}$ will not give the quality. To find the temperature corresponding to v' multiply s, the specific volume of one pound of saturated steam at pressure P, by the ratio $\frac{ab}{ac}$ as measured from the diagram. From superheated steam tables or by means of equation (311) determine the temperature corresponding to volume $s \times \frac{v'}{s}$ and pressure P. To transfer the point b to the temperature-entropy diagram draw the temperature line T' corresponding to that just determined and locate point b' on this line such that cb' = total entropy for pressure P and temperature T'. The total entropy may be taken from superheated steam tables

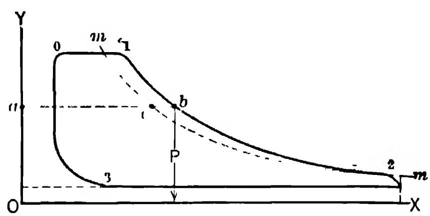


FIG. 706

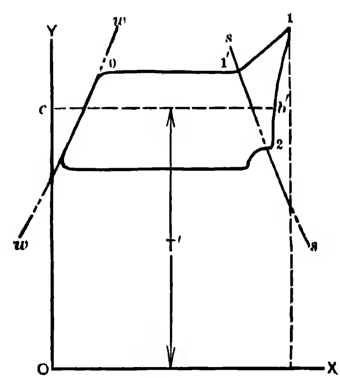


FIG. 707.

or it may be calculated from equation (342). For the problem under consideration the entropy thus obtained must be multiplied by 0.09, the weight of fluid expanding per cycle. The locus of the point b' will be the desired diagram.

402. Steam Accounted for by Indicator Diagrams at Points near Cut-off and Release. — The steam accounted for, expressed in pounds per i.hp. per hour, may readily be found by using the equation

$$\frac{13{,}750}{\text{m.e.p.}}[(C + E)\,W_c - (H + E)\,W_h], \tag{429}$$

in which

m.e.p. = mean effective pressure,

C = proportion of direct stroke completed at points on expansion line near cut-off or release,

E = proportion of clearance,

H = proportion of return stroke uncompleted at point on compression line just after exhaust closure.

W_c = weight of 1 cu. ft. steam at pressure shown at cut-off or release point,

W_h = weight of 1 cu. ft. steam at pressure shown at compression point.

The points near cut-off, release and compression referred to are indicated in Fig. 708.

In multiple expansion engines the mean effective pressure to be used in the above formula is the aggregate m.e.p. referred to the cylinder under consideration. In a compound engine the aggregate m.e.p. for the h-p. cylinder is the sum of the actual m.e.p. of the h-p. cylinder and that of the l-p. cylinder multiplied by the cylinder ratio. Likewise the aggregate m.e.p. for the l-p. cylinder is the sum of the actual m.e.p. of the l-p. cylinder and the m.e.p. of the h-p. cylinder divided by the cylinder ratio.

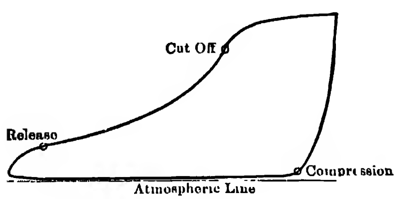


FIG. 708. Points where "Steam Accounted for by Indicator" is Computed.

The relation between the weight of steam shown by the indicator at any point in the expansion line and the weight of the mixture of steam and water in the cylinder may be represented graphically by plotting on the diagram a saturated steam curve showing the total consumption per stroke (including steam retained at compression) and comparing the abscissas of the curve with the abscissas of the expansion line, both measured from the line of no clearance.

403. Reheating Cycle. — This cycle has been employed for years in compound engines in which a reheater-receiver is placed between the high- and low-pressure cylinders. In some of the new turbine projects it is proposed to superheat the exhaust from the high-pressure unit in an auxiliary superheater placed inside the boiler before discharging it into the low-pressure element. The ideal temperature-entropy diagram for single-stage reheating is shown in Fig. 709. The portion of the curve *0123* is the same as for the Rankine cycle. From *3* the steam expands adiabatically to *4*. At *4* it is reheated at constant pressure to some point *5*, after which a second adiabatic expansion takes place from *5* to

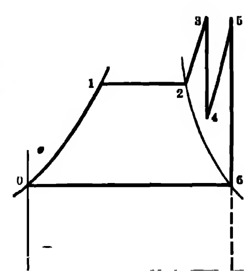


FIG. 709. Single-stage Reheating Cycle.

6. From *6* to *0* condensation of exhaust takes place at constant pressure just as in the Rankine cycle. The work done by a pound of steam in expanding to the reheating point *4* is $H_3 - H_4$, and in expanding from *5* to *6* is $H_5 - H_6$; the initial heat supplied is $H_3 - q_0$ and that during the process of reheating from *4* to *5*, $H_5 - H_4$. Total work done $= H_3 - H_4 + H_5 - H_6$; total heat supplied $= H_3 - q_0 + H_5 - H_4$, hence

$$\text{Eff.} = \frac{H_3 - H_4 + H_5 - H_6}{H_3 - H_4 + H_5 - q_0} \tag{429a}$$

in which

H = heat content of the steam and q = heat content of the liquid B.t.u. per lb. above 32 deg. fahr. for the state indicated by the subscript.

Example 112. — Calculate the efficiency of an ideal two-cylinder turbine, initial absolute pressure 600 lb. gage, initial temperature 750 deg. fahr., vacuum 1 in. abs., if the steam is exhausted from the high-pressure element at 185 lb. abs. pressure and is reheated to 750 deg. fahr. before passing into the low-pressure cylinder.

Solution. — From steam tables H_3 at 600 lb. and 750 deg. fahr. = 1378.7. From the Mollier diagram or by calculation, H_4, the heat content after adiabatic expansion from initial condition to 185 lb. pressure, is found to be 1247.6. From steam tables, H_6, heat content at 185 lb. pressure and 750 deg. temp., is 1401.1. From the Mollier diagram or by calculation, H_5, the heat content after adiabatic expansion from 185 lb. pressure and 750 deg. temp. to a back pressure of 1 in. of mercury is found to be 941.5. From steam tables, q_0, corresponding to 1 in. of mercury, is 47. Substituting these values in equation (429a) and solving

$$E_s = \frac{1737.8 - 1247.6 + 1401.1 - 941.5}{1737.8 - 1247.6 + 1401.1 - 47} = 0.398 \text{ or } 39.8 \text{ per cent.}$$

Figure 710 gives the temperature-entropy diagram for the two-stage reheating cycle. A glance at the curves in Fig. 328 will show that there is an appreciable gain in the efficiency of the two-stage cycle over the one-stage, but beyond two stages very little gain may be effected.

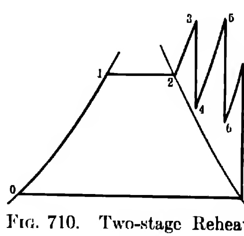


FIG. 710. Two-stage Reheating Cycle.

Reheating in Central Stations: Wohlenberg, Trans. A.S.M.E., Vol. 45, 1923.

High Pressure Reheating and Regenerating for Steam Power Plants: Hirshfeld and Ellenwood, Trans. A.S.M.E., Vol. 45, 1923.

The **Benson super-pressure** plant, while operating in the reheating cycle, differs from the conventional reheating plant in that steam is gener-

ated at the critical point (pressure 3200 lb. abs., temperature 706 deg. fahr.), throttled to 1500 lb. abs., reheated to a temperature of about 788 deg. fahr., and then passed through a high-pressure turbine exhausting at 200-lb. pressure. The exhaust is next reheated to about 662 deg. fahr. and then expanded in a standard turbine and condenser to a back pressure of 1-in. abs. The throttling process involves no heat loss and permits steam to be generated *without boiling*. This permits the use of small tubes without steam drums or chambers. The maximum thermal efficiency of a unit operating under the conditions specified above is about 42.5 per cent, assuming the values for the properties of steam as given in Marks and Davis' Steam Tables. Calculations for efficiency are the same as in the single-stage reheating cycle, equation (429*a*).

The Benson Super-pressure Plant: Power, May 22, 1923, p. 796; May 29, 1923, p. 842.

404. Regenerative Cycles. — In these cycles the condensate from the prime mover is passed through a series of feedwater heaters, the heating medium in which is steam bled from different pressure stages of the turbine or engine. With an infinite number of heaters the feedwater could be brought up to boiler temperature and the efficiency would be that of the Carnot cycle, as shown in paragraph 395. This is true only for saturated steam, since with superheated steam the maximum theoretical temperature of the condensate can never exceed that of saturated steam corresponding to steam at initial or throttle conditions and therefore the efficiency will be less than that of the Carnot cycle.

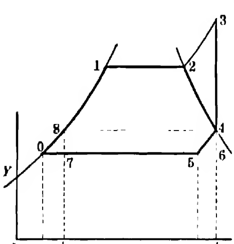


FIG. 711. Regenerative Cycle.

Figure 711 shows the temperature-entropy diagram of an ideal regenerative cycle with infinite number of stages in which the maximum temperature of the feedwater is taken as that of the liquid at a pressure, corresponding to the point in the expansion line where saturation begins, or in other words where superheat disappears. In this particular cycle, therefore, no superheated steam is bled. The line *01234* is the same as in the conventional Rankine cycle. Point *4* is the saturation point in the expansion line and represents the initial bleeding point. If the steam bled and that remaining in the turbine are mixed together the line *45* represents the condition of the mixture in passing from point *4* to the condenser. The line *50* represents the rejection of heat to the condenser. The line *45* is drawn parallel to *08* since the heat absorbed by the condensate *a08b* is equal to that absorbed from the bled steam *c54d*. The heat converted to work is represented by the area *012345* = area *a0123d* − area *a084d* +

area *7846* (= area *0845*), or, $H_3 - H_4 + (T_4 - T_0)\ (N_3 - \theta_4)$. The net heat supplied = area *b7812345c* = $H_3 - q_4$. Therefore the efficiency of the cycle is

$$\text{Eff.} = \frac{H_3 - H_4 + (T_4 - T_0)\ (N_3 - \theta_4)}{H_3 - q_4} \tag{429b}$$

in which

H = heat content of the steam, q = heat content of the liquid, T = absolute temperature, N = total entropy, and θ = entropy of the liquid for the state indicated by the subscript.

Example 113. — Calculate the thermal efficiency of a turbine working in the ideal regenerative cycle described above, if the conditions are as follows: Initial pressure, 600 lb. abs.; initial temperature, 750 deg. fahr.; back pressure, 1 in. mercury abs.

Solution. — From steam tables, H_3 at 600 lb. abs. and 750 deg. fahr. = 1378.7 B.t.u. per lb.; H_4, the heat content after expanding adiabatically from *3* to *4*, is found from the Mollier diagram or by calculation to be 1187.3, corresponding to a pressure of 94 lb. abs. From steam tables, $T_4 = 323.3 + 460 = 783.3$; $T_0 = 79 + 460 = 539$; N_3 at 600 lbs. and 750 deg. fahr. = 1.6094; θ_0 at 94 lb. abs. = 0.4677; $q_4 = 293.3$; $q_0 = 47$.

Substituting these values in equation (429*b*), and reducing, we have

$$\text{Eff.} = \frac{1378.7 - 1187.3 + (783.3 - 539)\ (1.6094 - 0.4677)}{1378.7 - 293.3} = 0.433,$$

or 43.3 per cent.

Figure 712 shows the temperature-entropy diagram of an ideal regenerative cycle with infinite number of stages, in which the maximum temperature of the feedwater is taken as that of the liquid at throttle conditions, *i.e.*, with steam initially superheated some of the bled steam will also be superheated. The equation for efficiency is the same as that of the preceding cycle; since the only difference lies in the point at which bleeding takes place.

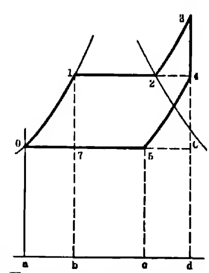


Fig. 712. Regenerative Cycle.

Example 114. — Using the data in Example 113, calculate the efficiency of the turbine when working on the above cycle.

Solution. — H_3 = 1378.7, as previously determined; H_4, the heat content after expanding adiabatically from *3* to *4* $= H_2 + T_4\ (N_3 - N_2)$. From steam tables H_2 and N_2 at 600 lb. and

saturation = 1199.8 and 1.4414, respectively; $T_4 = 486.5 + 460 = 946.5$; $T_0 = 79 + 460 = 539$; $N_3 = 1.6094$; $\theta_0 = 0.6679$; $q_4 = 468$; $q_0 = 47$.

$$H_4 = 1199.8 + 946.5\,(1.6094 - 1.4414) = 1358.8$$

Substituting these values in equation (429*b*) and reducing, we have

$$\text{Eff.} = \frac{1378.7 - 1358.8 + (946.5 - 539)\,(1.6094 - 0.6679)}{1378.7 - 468} = 0.443,$$

or 44.3 per cent.

It is impractical, from a constructive and operative standpoint, to have more than, say, three or four bleeding stages; therefore, the ideal cycles previously analyzed are of purely academic value. It is a comparatively simple matter to establish an ideal regenerative cycle for one or more stages which can be closely paralleled in practice. The efficiency of a single-stage cycle is readily calculated as follows:

Let H_1 = heat content at throttle conditions, B.t.u. per lb.

H_2 = heat content of the steam at the extraction point, after adiabatic expansion from throttle conditions.

H_n = heat content of the steam at exhaust pressure after expanding adiabatically from throttle conditions.

w = weight of steam bled, lb. per lb. of condensate.

q_2, q_n = heat of the liquid at bleeder and exhaust pressure, respectively.

Then, the work done by w lbs. of the bled steam = $(H_1 - H_2)w$ and $w = (q_2 - q_n) \div (H_2 - q_2)$.
Work done per 1 lb. of condensate = $H_1 - H_n$.
Heat added to $(1 + w)$ lb. of steam = $(1 + w)\,(H_1 - q_2)$.

$$\text{Eff.} = \frac{(H_1 - H_n) + w\,(H_1 - H_2)}{(1 + w)\,(H_1 - q_2)}. \tag{429c}$$

Example 115. — Using the data in Example 113, calculate the efficiency of a single-stage bleeder turbine if the steam is bled at the 25 lb. abs. stage.

Solution. — $H_1 = 1378.7$ as previously determined. H_2 for an adiabatic drop from 600 lb. abs. and 750 deg. fahr. to 25 lb. abs. = 108.6. H_n for an adiabatic drop from 600 lb. abs. and 750 deg. fahr. = 864. From steam tables $q_2 = 208$, $q_n = 47$. Then, $w = (208 - 47) \div (1086 - 208) = 0.182$. Substituting these values in equation (429*c*) and solving

$$\text{Eff.} = \frac{(1378.7 - 864) + 0.182\,(1378.7 - 1086)}{1.182\,(1378.7 - 208)} = 0.41.$$

Any number of stages may be analyzed in a similar manner.

This ideal efficiency cannot be realized in the actual single-bleeder stage turbine because of the turbine and generator losses, and the pressure drops and heat losses in the heater.

A Study of Steam Power Plant Cycles Hirshfeld and Ellenwood, Power Plant Engrg., Dec. 15, 1923, p. 1244; Trans. A.S.M.E., Vol. 45, 1923.

Economic Characteristics of Stage Feedwater Heating by Extraction: Brown and Drewry, Mech. Engrg., Mar., 1924, p. 118; Trans. A.S.M.E., Vol. 45, 1923

Feedwater Heating for High Thermal Efficiency: Helander, Trans A.S.M.E., Vol. 44, 1923, p. 1055.

The Commercial Economy of High Pressure and High Superheat in the Central Station: Orrok, Trans. A.S.M.E., Vol 44, 1922, p. 1119.

The Increase in Thermal Efficiency Due to Resuperheating in Steam Turbines. Blowney and Warren, Trans. A.S.M.E., Vol. 46, 1924.

405. Combined Reheater-Regenerative Cycle. — In some of the latest turbine projects it is proposed to use multi-cylinder turbines with reheating between cylinders and bleeding in the low-pressure cylinder. The thermal efficiency may be calculated by taking each step in the cycle and analyzing as outlined above.

The 50,000-kw. Parson turbine at the Crawford Avenue Station of the Commonwealth Edison Co., Chicago, Ill., is a notable example of the combined reheater-regenerative cycle. The turbine is a 3-cylinder unit and operates as follows: steam is generated at 600 lb. gage pressure, temperature 750 deg. fahr. and is supplied to the high pressure unit at a pressure of 550 lb. gage. It leaves the high-pressure cylinder at a pressure of 100 lb. gage and is reheated to 700 deg. fahr. before entering the intermediate cylinder. After leaving the latter at a pressure of 2 lb. gage, it enters the low-pressure cylinder and is exhausted into the condenser at a vacuum of 29.25 in. of Hg. About 22 per cent of the total heat entering the turbine is extracted at three points and raises the temperature of the feedwater from 65 to 315 deg. fahr. A heat consumption of 10,265 B.t.u. per kw-hr. is expected, corresponding to a thermal efficiency (steam to electricity) of 33.2 per cent.

CHAPTER XXIV. — SUPPLEMENTARY

PROPERTIES OF AIR. — DRY, SATURATED, AND PARTIALLY SATURATED

406. General. — Tables and charts giving the simultaneous physical and thermal properties of dry and saturated air for various temperatures are of great assistance in solving problems relative to the design and performance of evaporative surface condensers, water-cooling apparatus and air-conditioning devices. Table 138 gives the properties of dry and saturated air for various temperatures ranging from 0 to 212 deg. fahr. and Figs. 713 and 714 give a complete psychrometric chart for all conditions of dry, saturated, and partially saturated air within a temperature range of 20 to 350 deg. fahr. These charts are extremely useful in avoiding laborious calculations.

407. Dry Air. — The physical and thermal properties of dry air as used in these tables and charts are based on the following laws established by the latest experiments with gases and vapors:

$$\frac{P_a V_a}{T_a} = \text{constant} = 0.754, \tag{430}$$

$$C_{pa} = 0.2411 + 0.0000045\,(t_1 + t_2), \tag{431}$$

$$H_a = C_{pa}\,(t_2 - t_1), \tag{432}$$

in which

P_a = absolute pressure of the dry air, in. of mercury,
V_a = volume of 1 lb. of dry air, cu. ft.,
T_a = absolute temperature of the air, deg. fahr.,
C_{pa} = mean specific heat of air at constant pressure between temperatures t_1 and t_2,
H_a = heat content, B.t.u. per lb. of air above temperature t_1,
t_1 = initial temperature, deg. fahr.,
t_2 = final temperature, deg. fahr.

A sample calculation of the properties of dry air as listed in Table 138 is given in Example 116.

Example 116. — Required the specific volume and density of dry air at 100 deg. fahr. under standard atmospheric pressure (= 29.92 in.). Required also the heat content per lb. above 0 deg. fahr.

TABLE 138.

PROPERTIES OF SATURATED AIR. (Barometer 29.921.)

Mixture of Air Saturated with Water Vapor

Temperature, Degrees Fahr.	Weight of 1000 Cu. Ft of Dry Air, Pounds	Volume of One Lb. of Dry Air, Cu Ft.	Elastic Force of Vapor, In of Mercury *	Elastic Force of the Dry Air in the Mixture, In of Mercury.	Weight of 1000 Cu. Ft., Lb.		
					Weight of the Dry Air, Content	Weight of the Vapor, Content *	Total Weight of the Mixture
1	2	3	4	5	6	7	8
0	86.35	11.58	0.037	29.88	86.23	0.067	86.90
10	84.53	11.83	0.063	29.85	84.31	0.110	84.42
20	82.71	12.09	0.103	29.81	82.44	0.177	82.62
30	81.04	12.31	0.165	29.76	80.62	0.278	80.90
32	80.71	12.39	0.181	29.71	80.21	0.303	80.54
35	80.19	12.47	0.203	29.72	79.70	0.340	80.04
40	79.43	12.59	0.248	29.67	78.77	0.410	79.18
45	78.61	12.72	0.300	29.62	77.86	0.492	78.35
50	77.88	12.84	0.362	29.56	76.94	0.588	77.53
55	77.10	12.97	0.436	29.48	75.98	0.699	76.68
60	76.33	13.10	0.521	29.40	75.05	0.823	75.88
62	76.04	13.15	0.560	29.36	74.66	0.887	75.54
65	75.64	13.22	0.622	29.30	74.08	0.979	75.06
70	74.91	13.35	0.739	29.18	73.08	1.153	74.23
72	74.63	13.40	0.790	29.13	72.68	1.229	73.90
75	74.21	13.48	0.874	29.05	72.08	1.352	73.42
80	73.53	13.60	1.031	28.89	71.01	1.580	72.59
85	72.83	13.73	1.212	28.71	69.92	1.841	71.76
90	72.15	13.86	1.421	28.50	68.78	2.137	70.92
95	71.53	13.98	1.659	28.26	67.59	2.474	70.06
100	70.87	14.11	1.931	27.99	66.34	2.855	69.19
105	70.22	14.24	2.241	27.69	65.05	3.285	68.33
110	69.64	14.36	2.594	27.33	63.64	3.769	67.41
115	69.01	14.49	2.993	26.93	62.16	4.312	66.47
120	68.40	14.62	3.444	26.48	60.60	4.920	65.52
125	67.80	14.75	3.952	25.97	58.92	5.599	64.52
130	67.20	14.88	4.523	25.40	57.14	6.356	63.50
135	66.67	15.00	5.163	24.76	55.23	7.187	62.43
140	66.09	15.13	5.878	24.04	53.18	8.130	61.31
145	65.53	15.26	6.677	23.25	51.01	9.160	60.17
150	64.98	15.39	7.566	22.35	48.63	10.30	58.93
155	64.43	15.52	8.554	21.37	46.12	11.56	57.68
160	63.94	15.64	9.649	20.27	43.39	12.94	56.33
165	63.41	15.77	10.86	19.06	40.47	14.45	54.92
170	62.89	15.90	12.20	17.72	37.33	16.11	53.44
175	62.46	16.03	13.67	16.25	33.96	17.93	51.89
180	61.88	16.16	15.29	14.63	30.34	19.91	50.25
185	61.42	16.28	17.07	12.85	26.44	22.06	48.50
190	60.94	16.41	19.01	10.91	22.26	24.41	46.67
195	60.61	16.50	21.14	8.78	17.17	26.96	44.13
200	59.98	16.67	23.46	6.46	12.97	29.72	42.69
205	59.74	16.74	26.00	3.92	7.82	32.71	40.53
210	59.31	16.86	28.75	1.17	2.30	35.94	38.24
212	59.10	16.92	29.92	0	0	37.32	37.32

* Goodenough.

TABLE 138. — *Continued*

Temperature, Degrees Fahr.	Weight of Water Necessary to Saturate 100 Lb. of Dry Air	Volume of One Pound of Dry Air + Vapor to Saturate it, Cubic Feet	Heat Content per Pound of Dry Air, B.t.u.	Latent Heat of Vapor in One Lb. of Dry Air Saturated with Vapor, B.t.u.	Heat Content of One Lb. of Dry Air Saturated with Vapor, B.t.u.
0	0.078	11.59	0.000	0.964	0.964
10	0.131	11.86	2.411	1.608	4.019
20	0.214	12.13	4.823	2.623	7.446
30	0.344	12.41	7.234	4.195	11.429
32	0.378	12.47	7.716	4.058	11.783
35	0.427	12.55	8.44	4.57	13.02
40	0.520	12.70	9.65	5.56	15.21
45	0.632	12.85	10.86	6.73	17.59
50	0.764	13.00	12.07	8.12	20.19
55	0.920	13.1[illegible]	13.28	9.76	23.04
60	1.105	13.33	14.48	11.69	26.18
62	1.188	13.40	14.97	12.12	26.84
65	1.323	13.50	15.69	13.96	29.65
70	1.578	13.69	16.90	16.61	33.51
72	1.692	13.76	17.38	17.79	35.17
75	1.877	13.88	18.11	19.71	37.81
80	2.226	14.09	19.32	23.31	42.64
85	2.634	14.31	20.53	27.51	48.04
90	3.109	14.55	21.74	32.39	54.13
95	3.662	14.80	22.95	38.06	61.01
100	4.305	15.08	24.16	44.63	68.79
105	5.05	15.39	25.37	52.26	77.63
110	5.93	15.73	26.58	61.11	87.69
115	6.94	16.10	27.79	71.40	99.10
120	8.13	16.52	29.00	83.37	112.37
125	9.53	16.99	30.21	97.33	127.54
130	11.14	17.53	31.42	113.64	145.06
135	13.05	18.13	32.63	132.71	165.34
140	15.32	18.84	33.85	155.37	189.22
145	18.00	19.64	35.06	182.05	217.10
150	21.22	20.60	36.27	214.03	250.30
155	25.11	21.73	37.48	252.61	290.10
160	29.87	23.09	38.69	299.55	338.20
165	35.77	24.75	39.91	357.75	397.70
170	43.24	26.84	41.12	431.20	472.30
175	52.90	29.51	42.33	526.0	568.30
180	65.77	33.04	43.55	651.9	695.50
185	83.59	37.89	44.76	826.1	870.90
190	109.80	45.00	45.97	....	....
195	191.00	56.20	47.20	....	....
200	229.50	77.24	48.40	....	....
205	419.00	....	49.62	....	....
210	....	....	50.83	....	....
212	....	....	51.39	....	....

Solution. — From equation (430),

$$\frac{29.92 \times V_a}{100 + 459.6} = 0.754,$$

$$V_a = 14.11 \text{ cu. ft. per lb.}$$

$$\text{Density} = \frac{1}{14.11} = 0.071 \text{ lb. per cu. ft.}$$

From equation (431),

$$C_{pa} = 0.2411 + 0.0000045\ (0 + 100) = 0.2416,$$

and from equation (432),

$$H_a = 0.2416\ (100 - 0) = 24.16 \text{ B.t.u. per lb.}$$

408. Saturated Air. — Water, if placed in a vacuum chamber, will evaporate until the pressure in the chamber has reached that of vapor corresponding to the temperature of the water. If the water is introduced into a chamber containing dry air the evaporation will proceed precisely the same as in the vacuum until the pressure has risen by an amount corresponding to the vapor pressure for the temperature. In this case, according to Dalton's law (paragraph 216) each substance will exert the pressure it would if alone occupying the volume, and the final pressure will be the sum of that of the vapor and that of the air. Air is said to be saturated with moisture when it contains the saturated vapor of water. It might be better to say that the space is saturated since the presence of air has no effect on the vapor (the temperatures being the same) other than that the air retards the diffusion of water particles. Perfectly dry air does not exist in nature since evaporation of water from the earth's surface causes the atmosphere to be more or less diluted with vapor.

The weight of saturated water vapor per cubic foot depends only on the temperature and not on the presence of air.

The various properties for air completely saturated with water vapor may be calculated by means of equations (430) to (432), and Dalton's law, which may be expressed

$$P_a + P_v = P, \tag{433}$$

in which

P_a = absolute pressure of the dry air in the mixture, in. of mercury,
P_v = absolute pressure of saturated steam at the temperature of the mixture, in.,
P = total pressure, which for atmospheric conditions = 29.921.

Therefore,

$$P_a = P - P_v. \tag{434}$$

P_v may be taken directly from steam tables.

From equation (430),

$$V = V_a = \frac{0.754\ T_a}{P - P_v}, \tag{435}$$

in which

V_a = volume of 1 lb. of dry air (plus vapor to saturate) at pressure P_a and absolute temperature T_a,

V = volume of vapor in 1 lb. of dry air when saturated, cu. ft.

Evidently $$w_a = \frac{1}{V_a},$$

in which

w_a = weight of dry air in 1 cu. ft. of saturated mixture.

The weight, w_v, of vapor in 1 cu. ft. of saturated mixture is the density of saturated vapor at pressure P_v and temperature T_a. This may be taken directly from steam tables.

Total weight of mixture per cu. ft. $= w_a + w_v$.

The weight, w_v', of vapor necessary to saturate 1 lb. of dry air,

$$w_v' = Vw_v = V_a w_v. \tag{436}$$

Heat content H', or total heat in a mixture of 1 lb. of dry air saturated with water vapor, measured above 0 deg. fahr., and not including the heat of liquid, is

$$H' = C_{pa}t_a + r_v w_v', \tag{437}$$

in which

t_a = temperature of the mixture, deg. fahr.,

r_v = latent heat of saturated vapor at temperature t_a and pressure P_v.

An application of these formulas to the calculation of the various quantities in Table 138 for a temperature of 100 deg. fahr. is given in Example 117.

Example 117. — Required the following properties of atmospheric air completely saturated with water vapor when the temperature of the mixture is 100 deg. fahr.: Elastic force or pressure of the vapor and of the dry air in the mixture, volume of 1 lb. of dry air plus vapor to saturate it, weight of dry air and vapor in 1000 cu. ft. of mixture, weight of water necessary to saturate 100 lb. of dry air, latent heat of the vapor content of 1 lb. of mixture and the heat content of 1 lb. of dry air saturated with vapor.

6. From *6* to *0* condensation of exhaust takes place at constant pressure just as in the Rankine cycle. The work done by a pound of steam in expanding to the reheating point *4* is $H_3 - H_4$, and in expanding from *5* to *6* is $H_5 - H_6$; the initial heat supplied is $H_3 - q_0$ and that during the process of reheating from *4* to *5*, $H_5 - H_4$. Total work done $= H_3 - H_4 + H_5 - H_6$; total heat supplied $= H_3 - q_0 + H_5 - H_4$, hence

$$\text{Eff.} = \frac{H_3 - H_4 + H_5 - H_6}{H_3 - H_4 + H_5 - q_0} \tag{429a}$$

in which

H = heat content of the steam and q = heat content of the liquid B.t.u. per lb. above 32 deg. fahr. for the state indicated by the subscript.

Example 112. — Calculate the efficiency of an ideal two-cylinder turbine, initial absolute pressure 600 lb. gage, initial temperature 750 deg. fahr., vacuum 1 in. abs., if the steam is exhausted from the high-pressure element at 185 lb. abs. pressure and is reheated to 750 deg. fahr. before passing into the low-pressure cylinder.

Solution. — From steam tables H_3 at 600 lb. and 750 deg. fahr. = 1378.7. From the Mollier diagram or by calculation, H_4, the heat content after adiabatic expansion from initial condition to 185 lb. pressure, is found to be 1247.6. From steam tables, H_6, heat content at 185 lb. pressure and 750 deg. temp., is 1401.1. From the Mollier diagram or by calculation, H_5, the heat content after adiabatic expansion from 185 lb. pressure and 750 deg. temp. to a back pressure of 1 in. of mercury is found to be 941.5. From steam tables, q_0, corresponding to 1 in. of mercury, is 47. Substituting these values in equation (429a) and solving

$$E_s = \frac{1737.8 - 1247.6 + 1401.1 - 941.5}{1737.8 - 1247.6 + 1401.1 - 47} = 0.398 \text{ or } 39.8 \text{ per cent.}$$

Figure 710 gives the temperature-entropy diagram for the two-stage reheating cycle. A glance at the curves in Fig. 328 will show that there is an appreciable gain in the efficiency of the two-stage cycle over the one-stage, but beyond two stages very little gain may be effected.

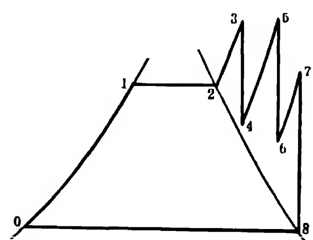


FIG. 710. Two-stage Reheating Cycle.

Reheating in Central Stations: Wohlenberg, Trans. A.S.M.E., Vol. 45, 1923.
High Pressure Reheating and Regenerating for Steam Power Plants: Hirshfeld and Ellenwood, Trans. A.S.M.E., Vol. 45, 1923.

The **Benson super-pressure** plant, while operating in the reheating cycle, differs from the conventional reheating plant in that steam is gener-

ated at the critical point (pressure 3200 lb. abs., temperature 706 deg. fahr.), throttled to 1500 lb. abs., reheated to a temperature of about 788 deg. fahr., and then passed through a high-pressure turbine exhausting at 200-lb. pressure. The exhaust is next reheated to about 662 deg. fahr. and then expanded in a standard turbine and condenser to a back pressure of 1-in. abs. The throttling process involves no heat loss and permits steam to be generated *without boiling*. This permits the use of small tubes without steam drums or chambers. The maximum thermal efficiency of a unit operating under the conditions specified above is about 42.5 per cent, assuming the values for the properties of steam as given in Marks and Davis' Steam Tables. Calculations for efficiency are the same as in the single-stage reheating cycle, equation (429*a*).

The Benson Super-pressure Plant: Power, May 22, 1923, p. 796; May 29, 1923, p. 842.

404. Regenerative Cycles. — In these cycles the condensate from the prime mover is passed through a series of feedwater heaters, the heating medium in which is steam bled from different pressure stages of the turbine or engine. With an infinite number of heaters the feedwater could be brought up to boiler temperature and the efficiency would be that of the Carnot cycle, as shown in paragraph 395. This is true only for saturated steam, since with superheated steam the maximum theoretical temperature of the condensate can never exceed that of saturated steam corresponding to steam at initial or throttle conditions and therefore the efficiency will be less than that of the Carnot cycle.

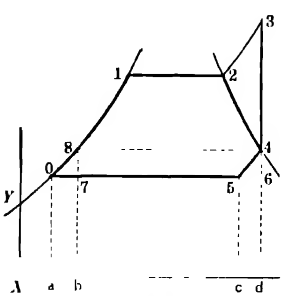


FIG. 711. Regenerative Cycle.

Figure 711 shows the temperature-entropy diagram of an ideal regenerative cycle with infinite number of stages in which the maximum temperature of the feedwater is taken as that of the liquid at a pressure, corresponding to the point in the expansion line where saturation begins, or in other words where superheat disappears. In this particular cycle, therefore, no superheated steam is bled. The line *01234* is the same as in the conventional Rankine cycle. Point *4* is the saturation point in the expansion line and represents the initial bleeding point. If the steam bled and that remaining in the turbine are mixed together the line *45* represents the condition of the mixture in passing from point *4* to the condenser. The line *50* represents the rejection of heat to the condenser. The line *45* is drawn parallel to *08* since the heat absorbed by the condensate *a08b* is equal to that absorbed from the bled steam *c54d*. The heat converted to work is represented by the area *012345* = area *a0123d* − area *a084d* +

area *7846* (= area *0845*), or, $H_3 - H_4 + (T_4 - T_0)(N_3 - \theta_4)$. The net heat supplied = area *b7812345c* = $H_3 - q_4$. Therefore the efficiency of the cycle is

$$\text{Eff.} = \frac{H_3 - H_4 + (T_4 - T_0)(N_3 - \theta_4)}{H_3 - q_4} \tag{429b}$$

in which

H = heat content of the steam, q = heat content of the liquid, T = absolute temperature, N = total entropy, and θ = entropy of the liquid for the state indicated by the subscript.

Example 113. — Calculate the thermal efficiency of a turbine working in the ideal regenerative cycle described above, if the conditions are as follows: Initial pressure, 600 lb. abs.; initial temperature, 750 deg. fahr.; back pressure, 1 in. mercury abs.

Solution. — From steam tables, H_3 at 600 lb. abs. and 750 deg. fahr. = 1378.7 B.t.u. per lb.; H_4, the heat content after expanding adiabatically from *3* to *4*, is found from the Mollier diagram or by calculation to be 1187.3, corresponding to a pressure of 94 lb. abs. From steam tables, $T_4 = 323.3 + 460 = 783.3$; $T_0 = 79 + 460 = 539$; N_3 at 600 lbs. and 750 deg. fahr. = 1.6094; θ_0 at 94 lb. abs. = 0.4677; $q_4 = 293.3$; $q_0 = 47$.

Substituting these values in equation (429*b*), and reducing, we have

$$\text{Eff.} = \frac{1378.7 - 1187.3 + (783.3 - 539)(1.6094 - 0.4677)}{1378.7 - 293.3} = 0.433,$$

or 43.3 per cent.

Figure 712 shows the temperature-entropy diagram of an ideal regenerative cycle with infinite number of stages, in which the maximum temperature of the feedwater is taken as that of the liquid at throttle conditions, *i.e.*, with steam initially superheated some of the bled steam will also be superheated. The equation for efficiency is the same as that of the preceding cycle; since the only difference lies in the point at which bleeding takes place.

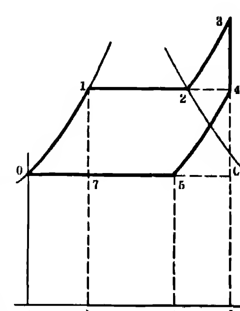


Fig. 712. Regenerative Cycle.

Example 114. — Using the data in Example 113, calculate the efficiency of the turbine when working on the above cycle.

Solution. — $H_3 = 1378.7$, as previously determined; H_4, the heat content after expanding adiabatically from *3* to *4* $= H_2 + T_4(N_3 - N_2)$. From steam tables H_2 and N_2 at 600 lb. and

saturation = 1199.8 and 1.4414, respectively; $T_4 = 486.5 + 460 = 946.5$; $T_0 = 79 + 460 = 539$; $N_3 = 1.6094$; $\theta_0 = 0.6679$; $q_4 = 468$; $q_0 = 47$.

$$H_4 = 1199.8 + 946.5\,(1.6094 - 1.4414) = 1358.8$$

Substituting these values in equation (429*b*) and reducing, we have

$$\text{Eff.} = \frac{1378.7 - 1358.8 + (946.5 - 539)\,(1.6094 - 0.6679)}{1378.7 - 468} = 0.443,$$

or 44.3 per cent.

It is impractical, from a constructive and operative standpoint, to have more than, say, three or four bleeding stages; therefore, the ideal cycles previously analyzed are of purely academic value. It is a comparatively simple matter to establish an ideal regenerative cycle for one or more stages which can be closely paralleled in practice. The efficiency of a single-stage cycle is readily calculated as follows:

Let H_1 = heat content at throttle conditions, B.t.u. per lb.

H_2 = heat content of the steam at the extraction point, after adiabatic expansion from throttle conditions.

H_n = heat content of the steam at exhaust pressure after expanding adiabatically from throttle conditions.

w = weight of steam bled, lb. per lb. of condensate.

q_2, q_n = heat of the liquid at bleeder and exhaust pressure, respectively.

Then, the work done by w lbs. of the bled steam $= (H_1 - H_2)w$ and $w = (q_2 - q_n) \div (H_2 - q_2)$.

Work done per 1 lb. of condensate $= H_1 - H_n$.

Heat added to $(1 + w)$ lb. of steam $= (1 + w)\,(H_1 - q_2)$.

$$\text{Eff.} = \frac{(H_1 - H_n) + w\,(H_1 - H_2)}{(1 + w)\,(H_1 - q_2)}. \qquad (429c)$$

Example 115. — Using the data in Example 113, calculate the efficiency of a single-stage bleeder turbine if the steam is bled at the 25 lb. abs. stage.

Solution. — $H_1 = 1378.7$ as previously determined. H_2 for an adiabatic drop from 600 lb. abs. and 750 deg. fahr. to 25 lb. abs. = 108.6. H_n for an adiabatic drop from 600 lb. abs. and 750 deg. fahr. = 864. From steam tables $q_2 = 208$, $q_n = 47$. Then, $w = (208 - 47) \div (1086 - 208) = 0.182$. Substituting these values in equation (429*c*) and solving

$$\text{Eff.} = \frac{(1378.7 - 864) + 0.182\,(1378.7 - 1086)}{1.182\,(1378.7 - 208)} = 0.41.$$

Any number of stages may be analyzed in a similar manner.

This ideal efficiency cannot be realized in the actual single-bleeder stage turbine because of the turbine and generator losses, and the pressure drops and heat losses in the heater.

A Study of Steam Power Plant Cycles Hirshfeld and Ellenwood, Power Plant Engrg., Dec. 15, 1923, p. 1244; Trans. A.S M.E., Vol. 45, 1923.

Economic Characteristics of Stage Feedwater Heating by Extraction Brown and Drewry, Mech. Engrg., Mar., 1924, p. 118; Trans. A.S M.E., Vol. 45, 1923.

Feedwater Heating for High Thermal Efficiency: Helander, Trans A.S.M.E., Vol. 44, 1923, p. 1055.

The Commercial Economy of High Pressure and High Superheat in the Central Station: Orrok, Trans. A.S.M.E., Vol 44, 1922, p. 1119.

The Increase in Thermal Efficiency Due to Resuperheating in Steam Turbines. Blowney and Warren, Trans. A.S.M.E., Vol. 46, 1924.

405. Combined Reheater-Regenerative Cycle. — In some of the latest turbine projects it is proposed to use multi-cylinder turbines with reheating between cylinders and bleeding in the low-pressure cylinder. The thermal efficiency may be calculated by taking each step in the cycle and analyzing as outlined above.

The 50,000-kw. Parson turbine at the Crawford Avenue Station of the Commonwealth Edison Co., Chicago, Ill., is a notable example of the combined reheater-regenerative cycle. The turbine is a 3-cylinder unit and operates as follows: steam is generated at 600 lb. gage pressure, temperature 750 deg. fahr. and is supplied to the high pressure unit at a pressure of 550 lb. gage. It leaves the high-pressure cylinder at a pressure of 100 lb. gage and is reheated to 700 deg. fahr. before entering the intermediate cylinder. After leaving the latter at a pressure of 2 lb. gage, it enters the low-pressure cylinder and is exhausted into the condenser at a vacuum of 29.25 in. of Hg. About 22 per cent of the total heat entering the turbine is extracted at three points and raises the temperature of the feedwater from 65 to 315 deg. fahr. A heat consumption of 10,265 B.t.u. per kw-hr. is expected, corresponding to a thermal efficiency (steam to electricity) of 33.2 per cent.

CHAPTER XXIV. — SUPPLEMENTARY

PROPERTIES OF AIR. — DRY, SATURATED, AND PARTIALLY SATURATED

406. General. — Tables and charts giving the simultaneous physical and thermal properties of dry and saturated air for various temperatures are of great assistance in solving problems relative to the design and performance of evaporative surface condensers, water-cooling apparatus and air-conditioning devices. Table 138 gives the properties of dry and saturated air for various temperatures ranging from 0 to 212 deg. fahr. and Figs. 713 and 714 give a complete psychrometric chart for all conditions of dry, saturated, and partially saturated air within a temperature range of 20 to 350 deg. fahr. These charts are extremely useful in avoiding laborious calculations.

407. Dry Air. — The physical and thermal properties of dry air as used in these tables and charts are based on the following laws established by the latest experiments with gases and vapors:

$$\frac{P_a V_a}{T_a} = \text{constant} = 0.754, \tag{430}$$

$$C_{pa} = 0.2411 + 0.0000045\,(t_1 + t_2), \tag{431}$$

$$H_a = C_{pa}\,(t_2 - t_1), \tag{432}$$

in which

P_a = absolute pressure of the dry air, in. of mercury,
V_a = volume of 1 lb. of dry air, cu. ft.,
T_a = absolute temperature of the air, deg. fahr.,
C_{pa} = mean specific heat of air at constant pressure between temperatures t_1 and t_2,
H_a = heat content, B.t.u. per lb. of air above temperature t_1,
t_1 = initial temperature, deg. fahr.,
t_2 = final temperature, deg. fahr.

A sample calculation of the properties of dry air as listed in Table 138 is given in Example 116.

Example 116. — Required the specific volume and density of dry air at 100 deg. fahr. under standard atmospheric pressure (= 29.92 in.). Required also the heat content per lb. above 0 deg. fahr.

TABLE 138.

PROPERTIES OF SATURATED AIR. (Barometer 29.921.)

Mixture of Air Saturated with Water Vapor

Temperature, Degrees Fahr.	Weight of 1000 Cu. Ft of Dry Air, Pounds	Volume of One Lb. of Dry Air, Cu Ft.	Elastic Force of Vapor, In of Mercury *	Elastic Force of the Dry Air in the Mixture, In. of Mercury.	Weight of 1000 Cu. Ft., Lb.		
					Weight of the Dry Air, Content	Weight of the Vapor, Content *	Total Weight of the Mixture
1	2	3	4	5	6	7	8
0	86.35	11.58	0.037	29.88	86.23	0.067	86.90
10	84.53	11.83	0.063	29.85	84.31	0.110	84.42
20	82.71	12.09	0.103	29.81	82.44	0.177	82.62
30	81.04	12.34	0.165	29.76	80.62	0.278	80.90
32	80.71	12.39	0.181	29.74	80.24	0.303	80.54
35	80.19	12.47	0.203	29.72	79.70	0.340	80.04
40	79.13	12.59	0.248	29.67	78.77	0.410	79.18
45	78.61	12.72	0.300	29.62	77.86	0.492	78.35
50	77.88	12.84	0.362	29.56	76.94	0.588	77.53
55	77.10	12.97	0.436	29.48	75.98	0.699	76.68
60	76.33	13.10	0.521	29.40	75.05	0.823	75.88
62	76.01	13.15	0.560	29.36	74.66	0.887	75.54
65	75.64	13.22	0.622	29.30	74.08	0.979	75.06
70	74.91	13.35	0.739	29.18	73.08	1.153	74.23
72	74.63	13.40	0.790	29.13	72.68	1.229	73.90
75	74.21	13.48	0.874	29.05	72.08	1.352	73.42
80	73.53	13.60	1.031	28.89	71.01	1.580	72.59
85	72.83	13.73	1.212	28.71	69.92	1.841	71.76
90	72.15	13.86	1.421	28.50	68.78	2.137	70.92
95	71.53	13.98	1.659	28.26	67.59	2.474	70.06
100	70.87	14.11	1.931	27.99	66.34	2.855	69.19
105	70.22	14.24	2.241	27.69	65.05	3.285	68.33
110	69.64	14.36	2.594	27.33	63.64	3.769	67.41
115	69.01	14.49	2.993	26.93	62.16	4.312	66.47
120	68.40	14.62	3.444	26.48	60.60	4.920	65.52
125	67.80	14.75	3.952	25.97	58.92	5.599	64.52
130	67.20	14.88	4.523	25.40	57.14	6.356	63.50
135	66.67	15.00	5.163	24.76	55.23	7.187	62.43
140	66.09	15.13	5.878	24.04	53.18	8.130	61.31
145	65.53	15.26	6.677	23.25	51.01	9.160	60.17
150	64.98	15.39	7.566	22.35	48.63	10.30	58.93
155	64.43	15.52	8.554	21.37	46.12	11.56	57.68
160	63.94	15.64	9.649	20.27	43.39	12.94	56.33
165	63.41	15.77	10.86	19.06	40.47	14.45	54.92
170	62.89	15.90	12.20	17.72	37.33	16.11	53.44
175	62.46	16.03	13.67	16.25	33.96	17.93	51.89
180	61.88	16.16	15.29	14.63	30.34	19.91	50.25
185	61.42	16.28	17.07	12.85	26.44	22.06	48.50
190	60.94	16.41	19.01	10.91	22.26	24.41	46.67
195	60.61	16.50	21.14	8.78	17.17	26.96	44.13
200	59.98	16.67	23.46	6.46	12.97	29.72	42.69
205	59.74	16.74	26.00	3.92	7.82	32.71	40.53
210	59.31	16.86	28.75	1.17	2.30	35.94	38.24
212	59.10	16.92	29.92	0	0	37.32	37.32

* Goodenough.

TABLE 138. — *Continued*

Temperature, Degrees Fahr.	Weight of Water Necessary to Saturate 100 Lb of Dry Air	Volume of One Pound of Dry Air + Vapor to Saturate it, Cubic Feet	Heat Content per Pound of Dry Air, B.t.u	Latent Heat of Vapor in One Lb of Dry Air Saturated with Vapor, B.t.u.	Heat Content of One Lb of Dry Air Saturated with Vapor, B.t.u.
0	0.078	11.59	0.000	0.964	0.964
10	0.131	11.86	2.411	1.608	4.019
20	0.214	12.13	4.823	2.623	7.446
30	0.344	12.41	7.234	4.195	11.429
32	0.378	12.47	7.716	4.058	11.783
35	0.427	12.55	8.44	4.57	13.02
40	0.520	12.70	9.65	5.56	15.21
45	0.632	12.85	10.86	6.73	17.59
50	0.764	13.00	12.07	8.12	20.19
55	0.920	13.15	13.28	9.76	23.04
60	1.105	13.33	14.48	11.69	26.18
62	1.188	13.40	14.97	12.12	26.84
65	1.323	13.50	15.69	13.96	29.65
70	1.578	13.69	16.90	16.61	33.51
72	1.692	13.76	17.38	17.79	35.17
75	1.877	13.88	18.11	19.71	37.81
80	2.226	14.09	19.32	23.31	42.64
85	2.634	14.31	20.53	27.51	48.04
90	3.109	14.55	21.74	32.39	54.13
95	3.662	14.80	22.95	38.06	61.01
100	4.305	15.08	24.16	44.63	68.79
105	5.05	15.39	25.37	52.26	77.63
110	5.93	15.73	26.58	61.11	87.69
115	6.94	16.10	27.79	71.40	99.10
120	8.13	16.52	29.00	83.37	112.37
125	9.53	16.99	30.21	97.33	127.54
130	11.14	17.53	31.42	113.64	145.06
135	13.05	18.13	32.63	132.71	165.34
140	15.32	18.84	33.85	155.37	189.22
145	18.00	19.64	35.06	182.05	217.10
150	21.22	20.60	36.27	214.03	250.30
155	25.11	21.73	37.48	252.61	290.10
160	29.87	23.09	38.69	299.55	338.20
165	35.77	24.75	39.91	357.75	397.70
170	43.24	26.84	41.12	431.20	472.30
175	52.90	29.51	42.33	526.0	568.30
180	65.77	33.04	43.55	651.9	695.50
185	83.59	37.89	44.76	826.1	870.90
190	109.80	45.00	45.97	....	....
195	191.00	56.20	47.20	....	....
200	229.50	77.24	48.40	....	....
205	419.00	....	49.62	....	....
210	....	....	50.83	....	....
212	....	....	51.39	....	....

Solution. — From equation (430),

$$\frac{29.92 \times V_a}{100 + 459.6} = 0.754,$$

$$V_a = 14.11 \text{ cu. ft. per lb.}$$

$$\text{Density} = \frac{1}{14.11} = 0.071 \text{ lb. per cu. ft.}$$

From equation (431),

$$C_{pa} = 0.2411 + 0.0000045\ (0 + 100) = 0.2416,$$

and from equation (432),

$$H_a = 0.2416\ (100 - 0) = 24.16 \text{ B.t.u. per lb.}$$

408. Saturated Air. — Water, if placed in a vacuum chamber, will evaporate until the pressure in the chamber has reached that of vapor corresponding to the temperature of the water. If the water is introduced into a chamber containing dry air the evaporation will proceed precisely the same as in the vacuum until the pressure has risen by an amount corresponding to the vapor pressure for the temperature. In this case, according to Dalton's law (paragraph 216) each substance will exert the pressure it would if alone occupying the volume, and the final pressure will be the sum of that of the vapor and that of the air. Air is said to be saturated with moisture when it contains the saturated vapor of water. It might be better to say that the space is saturated since the presence of air has no effect on the vapor (the temperatures being the same) other than that the air retards the diffusion of water particles. Perfectly dry air does not exist in nature since evaporation of water from the earth's surface causes the atmosphere to be more or less diluted with vapor.

The weight of saturated water vapor per cubic foot depends only on the temperature and not on the presence of air.

The various properties for air completely saturated with water vapor may be calculated by means of equations (430) to (432), and Dalton's law, which may be expressed

$$P_a + P_v = P, \tag{433}$$

in which

P_a = absolute pressure of the dry air in the mixture, in. of mercury,
P_v = absolute pressure of saturated steam at the temperature of the mixture, in.,
P = total pressure, which for atmospheric conditions = 29.921.

Therefore,

$$P_a = P - P_v. \tag{434}$$

P_v may be taken directly from steam tables.

From equation (430),

$$V = V_a = \frac{0.754\ T_a}{P - P_v}, \tag{435}$$

in which

V_a = volume of 1 lb. of dry air (plus vapor to saturate) at pressure P_a and absolute temperature T_a,

V = volume of vapor in 1 lb. of dry air when saturated, cu. ft.

Evidently
$$w_a = \frac{1}{V_a},$$

in which

w_a = weight of dry air in 1 cu. ft. of saturated mixture.

The weight, w_v, of vapor in 1 cu. ft. of saturated mixture is the density of saturated vapor at pressure P_v and temperature T_a. This may be taken directly from steam tables.

Total weight of mixture per cu. ft. = $w_a + w_v$.

The weight, w_v', of vapor necessary to saturate 1 lb. of dry air,

$$w_v' = Vw_v = V_a w_v. \tag{436}$$

Heat content H', or total heat in a mixture of 1 lb. of dry air saturated with water vapor, measured above 0 deg. fahr., and not including the heat of liquid, is

$$H' = C_{pa}t_a + r_v w_v', \tag{437}$$

in which

t_a = temperature of the mixture, deg. fahr.,

r_v = latent heat of saturated vapor at temperature t_a and pressure P_v.

An application of these formulas to the calculation of the various quantities in Table 138 for a temperature of 100 deg. fahr. is given in Example 117.

Example 117. — Required the following properties of atmospheric air completely saturated with water vapor when the temperature of the mixture is 100 deg. fahr.: Elastic force or pressure of the vapor and of the dry air in the mixture, volume of 1 lb. of dry air plus vapor to saturate it, weight of dry air and vapor in 1000 cu. ft. of mixture, weight of water necessary to saturate 100 lb. of dry air, latent heat of the vapor content of 1 lb. of mixture and the heat content of 1 lb. of dry air saturated with vapor.

Solution. — Pressure of vapor in the mixture:

$$P_v = 1.931 \text{ in. (from steam tables).}$$

Pressure of dry air in the mixture:

$$\begin{aligned} P_a &= P - P_v \\ &= 29.921 - 1.931 = 27.99 \text{ in.} \end{aligned}$$

Volume of 1 lb. of dry air saturated with vapor:

$$\begin{aligned} V_a &= \frac{0.754\ T_a}{P - P_v} \\ &\frac{0.754\ (100 + 459.6)}{29.921 - 1.931} = 15.08 \text{ cu. ft.} \end{aligned}$$

Weight of dry air in 1000 cu. ft. of saturated mixture:

$$w_a = \frac{1}{V_a} = \frac{1}{15.08} = 0.06634 \text{ lb. per cu. ft.}$$

$$1000\ w_a = 1000 \times 0.06634 = 66.34 \text{ lb.}$$

Weight of water vapor in 1000 cu. ft. of mixture:

$$\begin{aligned} w_v &= 0.002855 \text{ lb. per cu. ft. (from steam tables),} \\ 1000\ w_v &= 1000 \times 0.002855 = 2.855 \text{ lb.} \end{aligned}$$

Total weight of 1000 cu. ft. of mixture:

$$= 66.34 + 2.855 = 69.19+ \text{ lb.}$$

Weight of vapor necessary to saturate 100 lb. of dry air:

$$\begin{aligned} w_v' &= Vw_v = V_a w_v \\ &= 15.08 \times 0.002855 = 0.04305 \text{ lb. per lb. of dry air} \\ &= 0.04305 \times 100 = 4.305 \text{ lb. per 100 lb. of dry air.} \end{aligned}$$

Total heat of the dry air content, above 0 deg. fahr.:

$$\begin{aligned} H_a &= C'_{pa}\ (100 - 0) \\ &= 0.2416 \times 100 = 24.16 \text{ B.t.u. per lb.} \end{aligned}$$

Latent heat of the vapor content:

$$r_v w_v' = 1036.6 \times 0.04305 = 44.63 \text{ B.t.u.}$$

Total heat of 1 lb. of dry air saturated with vapor:

$$\begin{aligned} H_1 &= H_a + r_v w_v', \\ &= 24.16 + 44.63 = 68.79 \text{ B.t.u.} \end{aligned}$$

409. Partially Saturated Air. — As previously stated, air is said to be saturated with moisture when it contains the saturated vapor of water. In this condition the weight of vapor per cu. ft. corresponds to the density of saturated steam at the temperature of the mixture. If the body of air contains only a fraction of the weight of vapor corresponding to saturation it is said to be partially saturated and the fraction is called the

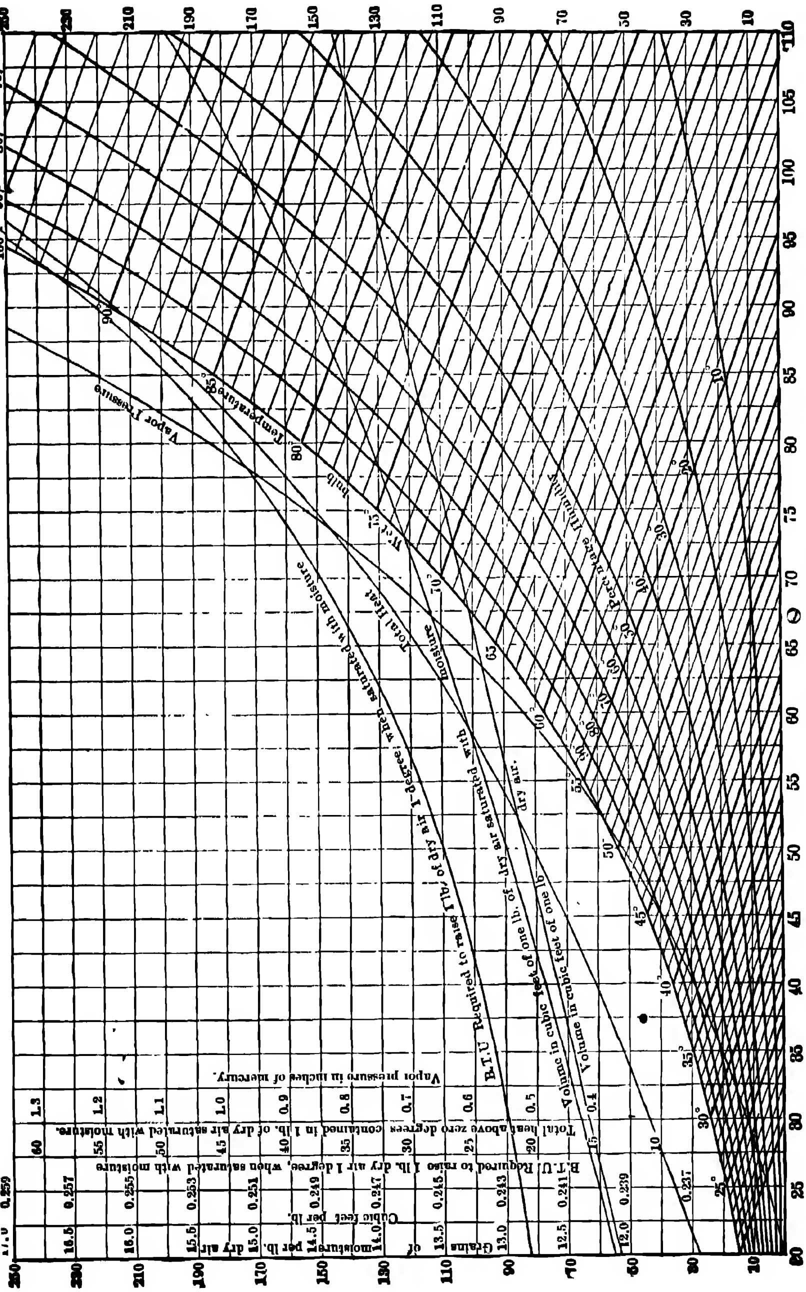


FIG. 713. Psychrometric Chart (W. H. Carrier).

Weight of vapor in 1 lb. of dry air:

From equation (439)

$$w = \frac{1047.4 \times 0.02226 - 0.2416 \times 20}{1047.4 + 0.429 \times 20} = 0.0175 \text{ lb.}$$

w may also be closely approximated as follows:

$$w = hV \times \text{density of saturated vapor at 100 deg. fahr.}$$
$$= 0.42 \times 14.5 \times 0.002855 = 0.0174 \text{ lb.}$$

Total heat in 1 lb. of dry air containing w lb. of vapor at temperature t_d:

From equation (443)

$$H' = 1047.4 \times 0.02226 + 0.2415 \times 80 = 42.46 \text{ B.t.u.}$$

H' may also be approximated from the values in Table 138.

H' = heat content of the dry air + h × latent heat content of saturated vapor at temperature t_d = 100
= 24.16 + 0.42 × 44.63 = 42.9 B.t.u.

An application of Table 138 and the psychrometric charts in Fig. 713 is given in Examples 119 and 120.

Example 119. — Atmospheric air at 40 deg. fahr. and relative humidity 0.80 is to be conditioned to 70 deg. fahr. and relative humidity 0.50. Determine the amount of moisture and heat to be added, (1) by means of Table 138 and (2) by means of the curves in Fig. 713.

Solution. —

From Table 138:

Original moisture content = 0.52 × 0.8 = 0.416 lb. per 100 lb. of dry air.

Final moisture content = 1.578 × 0.5 = 0.789 lb. per 100 lb. of dry air.

Moisture to be added = 0.789 − 0.416 = 0.373 lb. per 100 lb. of dry air.

From Fig. 713:

Initial moisture content (intersection of t_d = 40 and h = 80 per cent) = 29 grains per lb. of dry air.

Final moisture content (intersection of t_d = 70 and h = 50 per cent) = 55 grains per lb. of dry air.

Moisture to be added $= 100\left(\frac{55 - 29}{7000}\right) = 0.371$ lb. per 100 lb. of dry air. (7000 = grains per lb.)

From Table 138:

Initial heat content = 9.65 + 0.8 × 5.56 = 14.1 B.t.u. per lb.
Final heat content = 16.90 + 0.5 × 16.61 = 25.20 B.t.u. per lb.
Heat required = 25.20 − 14.1 = 11.10 B.t.u. per lb. of dry air.

From Fig. 713:

Initial heat content (intersection of $t_d = 40$ and $h = 80$ per cent) gives wet bulb $t_w = 37.5$; follow constant temperature line $t_w = 37.5$ until it intersects saturation line $h = 100$ per cent; trace vertically upward to intersection of "total heat" line and read from marginal notation 14.1 B.t.u. per lb.

Final heat content (intersection of $t_d = 70$ and $h = 50$ per cent) gives $t_w = 58.5$; follow constant temperature line $t_w = 58.5$ until it intersects line $h = 100$ per cent; trace vertically upward to intersection of "total heat" line and read 25.2.

The charts in Figs. 713 and 714 are reproduced to a greatly reduced scale and the readings cannot be made with the accuracy indicated in the example. In the original charts the wet- and dry-bulb temperature can be read to an accuracy of 0.1 degree and the other quantities proportionately.

Example 120. — Atmospheric air at 90 deg. fahr. and relative humidity of 80 per cent is to be conditioned to 70 deg. fahr. and 50 per cent relative humidity. Determine the temperature to which the original mixture must be reduced in order to have a relative humidity of 50 per cent when heated to 70 deg. fahr. Determine also the amount of heat to be abstracted to effect the initial cooling and that to be supplied to bring it to the final desired condition.

Solution. — Moisture content at $t_d = 90$ and $h = 0.8 = 3.109 \times 0.8 = 2.487$ lb. per 100 lb. of dry air, corresponding dew point = 83 deg. fahr., that is, at 83 deg. fahr. condensation begins.

Moisture content at $t_d = 70$ and $h = 0.5 = 1.578 \times 0.5 = 0.789$ lb. per 100 lb. of dry air. Corresponding dew point = 51.8 deg. fahr. This is the temperature to which the air must be cooled in order to have the required humidity when reheated to 70 deg. fahr. Heat content at $t_d = 90$ and $h = 0.8 = 21.74 + 0.8 \times 32.39 = 47.65$ B.t.u. per lb.

Heat content at $t_d = 51.8$ and $h = 1.0 = 21.19$ B.t.u. per lb.

Heat to be removed from water condensed due to cooling from 83 to 51.8 deg. fahr. $= \frac{2.487 - 0.789}{100} \times \frac{83 - 51.8}{2} = 0.26$ B.t.u. per lb.

(This is comparatively small and may be omitted.)

Total heat to be removed in cooling from initial conditions to 51.8 deg. fahr. $= 47.65 - (21.19 + 0.26) = 26.20$ B.t.u. per lb.

Heat content at $t_d = 70$ and $h = 0.5 = 16.9 + 0.5 \times 13.96 = 23.88$ B.t.u.

Heat to be added to retemper from 51.8 to 70 deg. $= 23.88 - 21.19 = 2.69$ B.t.u. per lb.

These values, neglecting the heat of the liquid, may be taken directly from the curves in Fig. 713 as shown in the preceding example.

Example 121. *Evaporative Surface Condenser.* — How many cubic feet of air and how many pounds of water spray must be forced through

From Table 138:

$$w_1 \text{ for dew point } 69.0 = 0.0152 \text{ lb.}$$
$$w_2 \text{ for dew point } 119.2 = 0.0793 \text{ lb.}$$

Moisture absorbed per lb. of dry air = 0.0793 − 0.0152 = 0.0641 lb.
Since the heat content or total heat is constant for a given wet-bulb temperature

$$H_1 \text{ for wet bulb } 72.2 = 35.3 \text{ B.t.u.}$$
$$H_2 \text{ for wet bulb } 119.2 = 110.5.$$

Heat absorbed per lb. of dry air and its vapor content.

$$H_2 - H = 110.5 - 35.3 = 75.2 \text{ B.t.u. per lb.}$$

These results check substantially with the calculated data.

Example 122. — Determine the quantity of air passing through the cooling tower and the weight of circulating water lost by evaporation in a surface-condensing power plant operating under the following conditions: Turbines, average load 1000 kw.; average water rate 20 lb. per kw-hr.; initial steam pressure 150 lb. abs.; superheat 50 deg. fahr.; vacuum 26.92 in.; barometer 29.92 in.; temperature of injection water, discharge water and outside air, 70, 100, and 65 deg. fahr., respectively; temperature of air leaving tower 90 deg. fahr.; wet bulb temperature of outside air and air leaving cooling tower 57 and 89 deg. fahr., respectively.

Solution. — Total heat to be abstracted from the steam =

$$1000 \times 20 \left(1223 - \frac{3412}{20} - 105^* + 32\right) = 19{,}580{,}000 \text{ B.t.u. per hr.}$$

* Assumed hotwell temperature.

Atmospheric air entering tower:

From the curves in Fig. 713 (dry-bulb temperature 65 deg. fahr. and wet bulb temperature 57 deg. fahr.).
Moisture content of 1 lb. of dry air, w_1 = 56 grains.
Total heat of 1 lb. of dry air, with its vapor content,

$$H_1 = 24.3 \text{ B.t.u.}$$

Air-vapor mixture leaving tower:

From the curves in Fig. 713 (dry-bulb 90 and wet-bulb 89).
Moisture content of 1 lb. of dry air, w_2 = 209 grains.
Total heat of 1 lb. of dry air, with its vapor content,

$$H_2 = 52.8 \text{ B.t.u.}$$

Moisture absorbed by 1 lb. of dry air in passing through the tower

$$= w_2 - w = 209 - 56 = 153 \text{ grains or } 0.02186 \text{ lb.}$$

Heat absorbed by 1 lb. of dry air (plus its initial vapor content) in passing through the tower

$$= H_2 - H = 52.8 - 24.3 = 28.5 \text{ B.t.u.}$$

Total weight of dry air required to abstract the heat from the circulating water

$$= \frac{19,580,000}{28.5} = 687,000 \text{ lb. per hr.}$$

Volume of 1 lb. of dry air and its vapor content entering tower

$$= 0.754\left(\frac{459.6 + 65}{29.54}\right) = 13.39 \text{ cu. ft.}$$

(29.54 = pressure of the dry air in the mixture $= 29.92 - 0.61 \times 0.6218$; 0.61 = relative humidity and 0.6218 = pressure of saturated vapor at 65 deg. fahr.)

Total volume of atmospheric air entering tower

$$= \frac{687,000 \times 13.39}{60} = 153,000 \text{ cu. ft. per min.}$$

The Temperatures of Evaporation of Water into Air: Carrier and Lindsay, Trans. A.S.M.E., Vol. 46, 1924.

The Design of Cooling Towers: Robinson, Trans. A.S.M.E., Vol. 44, 1922, p 669.

The Evaporation of a Liquid into a Gas: Lewis, Trans. A.S.M.E., Vol. 44, 1922, p. 325.

APPENDIX A

EQUIVALENT VALUES OF ELECTRICAL AND MECHANICAL UNITS

1 Myriawatt =

10 kilowatts
10,000 watts
13.41 horsepower
13.597 cheval-vapeur
13.597 pferde-kraft
26,552,000 foot pounds per hour
8,605,000 gram calories per hour
3,670,000 kilogram meters per hour
34,150 B.t.u. per hour
1.02 boiler horsepower

1 Horsepower =

745.7 watts
0.7457 kilowatt
0.07457 myriawatt
1.0139 cheval-vapeur
1.0139 pferde-kraft
33,000 foot pounds per minute
641,700 gram calories per hour
273,743 kilogram meters per hour
2,547 B.t.u. per hour

1 Joule =

1 watt second
0.10197 kilogram meter
0.73756 foot pound
0.239 gram calorie
0.0009486 B.t.u.

1 B.t.u. =

1054 watt seconds
777.5 foot pounds
107.5 kilogram meters
0.0003927 horsepower hour

1 Kilowatt =

0.1 myriawatt
1000 watts
1.341 horsepower
1.3597 cheval-vapeur
1.3597 pferde-kraft
2,655,200 foot pounds per hour
860,500 gram calories per hour
367,000 kilogram meters per hour
3,415 B.t.u. per hour
0 102 boiler horsepower

1 Cheval-Vapeur or Pferde-Kraft =

75 kilogram meters per second
0.07354 myriawatt
0.7357 kilowatt
0.9863 horsepower
32,550 foot pounds per minute
632,900 gram calories per hour
2,512 B.t.u. per hour

1 Foot Pound =

1.3558 joules
0.13826 kilogram meter
0.001286 B.t.u.
0.03241 gram calorie
0.000000505 horsepower hour

1 Kilogram-Meter =

7.233 foot pounds
9.806 joules
2.344 gram calories
0.0093 B.t.u.

APPENDIX B

MISCELLANEOUS CONVERSION FACTORS

1 POUND PER SQUARE INCH =

2.0355	inches of mercury at 32° F.
2.0416	inches of mercury at 62° F.
2.309	feet of water at 62° F.
0.07031	kilogram per square centimeter
0.06804	atmosphere
51.7	millimeters of mercury at 32° F.

1 FOOT OF WATER AT 62° F. =

0.433	pound per square inch
62.355	pounds per square foot
0.883	inch of mercury at 62° F.
821.2	feet of air at 62° F. and barometer 29.92

1 INCH OF WATER 62° F. =

0.0361	pound per square inch
5.196	pounds per square foot
0.5776	ounce per square inch
0.0735	inch of mercury at 62° F.
68.44	feet of air at 62° F. and barometer 29.92

1 FOOT OF AIR AT 32° F. AND BAROMETER 29.92 =

0.0761	pound per square foot
0.0146	inch of water at 62° F.

1 INCH OF MERCURY AT 62° F. =

0.4912	pound per square inch
1.134	feet of water at 62° F.
13.61	inches of water at 62° F.

1 ATMOSPHERE =

760.0	millimeters of mercury at 32° F.
14.7	pounds per square inch
29.921	inches of mercury at 32° F.
2116.0	pounds per square foot
1.033	kilograms per square centimeter
33.947	feet of water at 62° F.

1 MILLIMETER = 0 03937 inch

1 CENTIMETER = 0.3937 inch

1 METER = 39.37 inches

1 METER = 3.2808 feet

1 SQUARE METER = 10.764 square feet

1 LITER =

61.023 cubic inches
0.264 U. S. gallons

1 GRAM =

1	cubic centimeter of distilled water
15.43	grains troy
0.0353	ounce

1 KILOGRAM =

2.20462 pounds avoirdupois

APPENDIX C

REFERENCES TO DETAILED DESCRIPTIONS OF MODERN CENTRAL AND ISOLATED STATIONS

CENTRAL STATIONS

Station	*Company*	*Reference*
Barbadoes Island	Counties Gas & Electric	Power Plant Engrg., Oct. 1, '23, p. 965.
Battle Creek	Consumers Power Co.	Power, June 6, '22, p. 881.
Cahokia	Union Elec. Lt. Co.	Power, Apr. 1, '24, p. 514.
		Power Plant Engrg., July 15, '23, p. 135.
•Calumet	Commonwealth Edison Co.	Power Plant Engrg., May 15, '22, p. 498.
		Power, May 30, '22, p. 842.
Colfax	Duquesne Lt. Co.	Power, May 24, '21, p. 824.
		Trans. A.S.M.E., Vol. 44, '22, p. 250.
Conners Creek	Detroit Edison.	Trans. A.S.M.E., Vol. 44, '22, p. 1033.
Crawford Ave.	Commonwealth Edison Co.	Power, July 15, '24, p. 88; Nov. 4, '24, p. 728; June 16, '25, p. 936.
Delaware	Ohio Elec.	Power Plant Engrg., May 24, '21, p. 806.
Devon	Conn. Lt. and Power	Power, Mar. 25, 1924, p. 474.
		Elec. Wld., Mar. 22, 1924, p. 563.
Hell Gate	United Elec. Lt.	Power, May 2, '22, p. 678.
		Power Plant Engrg., June 1, '23, p. 556.
Long Beach	So. Calif. Ed. C.	Power, Feb. 17, 1925, p. 246.
High Bridge	Northern States Power Co	Power, Dec. 23, '24, p. 1006
Hudson Ave.	Brooklyn Ed.	Power, May 13, '24, p. 750.
Kearney	Pub. Serv. El. Co., N. J	Power Plant Engrg., Mar. 1, '24, p. 311.
Lakeside	Milwaukee Elec. Ry.	Elec. Wld., Apr. 15, '22, p. 721.
		Power, Apr. 18, '22, p. 604.
		Power, Dec. 19, '22, p. 968.
Marysville	Detroit Edison Co.	Power, May 29, '23, p. 824.
Miami Fort.	Columbia Power Co.	Elec. Wld., Dec. 20, '24, p. 1305.
Middletown.	Met. Ed. Pa.	Power, Dec. 25, '23, p. 1025.
Northeast	Kan. City Lt. & Power.	Power, July 3, '23, p. 2.
		Power, Aug. 28, '23, p. 318.
Saginaw River	Consumers Power Co.	Power, July 22, '24, p. 122.
Seward	Penn. Pub. Serv.	Power, Aug. 23, '21, p. 278.
Somerset	Montaup Elec. Co	Power Plant Engrg. Apr. 1, 1925, p. 368.
Springdale	West Penn. Power Co.	Power, Sept. 28, '20, p. 488.
Steel Point	United Ill. Co., Conn.	Power, Feb. 12, '24, p. 238.
Waukegan	Pub. Serv. North. Ill.	Power, Jan. 15, '24, p. 80;
		Power Plant Engrg., Jan. 15, '24, p. 120.
West Reading	Met. Ed. Pa.	Power Plant Engrg., Aug. 1, '23, p. 757.
Weymouth	Edison Elec. Ill. Co.	Power, July 10, '23, p. 42.

INDUSTRIAL STATIONS

Ashtabula	Ashtabula Steel Co.	Power Plant Engrg., Aug. 15, '23, p. 808.
Baltimore, Md.	Am. Sup. Ref. Co.	Power, Sept. 12, '22, p. 441.
Chicago, Ill.	Great Western Laundry	Power, Jan. 29, '24, p. 163.
Cincinnati, O.	American Can Co.	Power, Nov. 11, '24, p. 755.
Detroit, Mich.	Dodge Bros.	Power, May 24, '21, p. 841.
Elizabeth, N. J.	Durant Motors	Power, Oct. 23, '23, p. 638.
Ensley Works	Tenn. Coal, Iron, R.R.	Power, Nov. 6, '23, p. 718.
Erie, Pa.	Hammermill Paper Co.	Power, Feb. 15, '21, p. 201.
Kokomo, Ind.	Pittsburgh Plate Glass Co.	Power, Oct. 31, '22, p. 675.
Locust Point, Md.	Am. Sup. Refin. Co.	Power, July 4, '22, p. 2.
Maurer, N. J.	Barber Asphalt	Power Plant Engrg., July 15, '23, p. 709.
Milwaukee, Wis. (P. C.)	Eline's Inc.	Power, Dec. 5, 1922, p. 869.
Newark, N. J.	Duratex	Power, Sept. 5, '22, p. 359.
Newark, N. J.	Clark Thread Co.	Power, Nov. 4, '24, p. 715.
North Canton, O.	Hoover Suction Sweeper Co.	Power, Nov. 14, '22, p. 751.
Olean, N. Y.	Vacuum Oil Co.	Power Plant Engrg., Nov. 1, '24, p. 715.
Point Breeze	Atlantic Refining	Power Plant Engrg., Aug. 1, '22, p. 739.
River Rouge	Ford Motor Co.	Power, Sept. 6, '21, p. 348.
St. Louis, Mo.	Bridge & Beach Co.	Power Plant Engrg., Mar. 1, '24, p. 267.
Toronto, Ohio	Follansbee Bros.	Power, July 25, '22, p. 117.

BUILDINGS

Boston, Mass.	Harvard Medical School.	Power, Apr. 1, '22, p. 349.
Chicago, Ill.	Field Museum	Power, Sept. 26, '22, p. 482.
Cleveland, O.	Federal Reserve Bank	Power Plant Engrg., Nov. 1, '23, p. 1064.
Jersey City	First Nat'l Bank	Power, Aug. 29, '22, p. 311.
New York	R. H. Macy	Power Plant Engrg., Dec. 15, '23, p. 216.
Detroit, Mich.	Book — Cadillac Hotel	Power, Jan. 19, 1926, p. 86.

an evaporative surface condenser of the fan type in order to condense 1000 lb. of steam per hour and maintain a vacuum of 25 in., barometer 29? (Atmospheric air 80 deg. fahr., relative humidity 70 per cent.) The air and vapor issue from the discharge pipe under pressure of 4 in. of water, temperature 120 deg. fahr., relative humidity 98 per cent.

Solution. — The absolute pressure in the condenser is $29.0 - 25.0 = 4$ in. of mercury.

The total heat to be withdrawn in order to cool and condense 1000 lb. of steam per hour at absolute pressure of 4 in. to 120 deg. fahr. is

$$1000\,[1114.8 - (120 - 32)] = 1{,}026{,}000 \text{ B.t.u.}$$

Neglecting radiation and leakage losses, this is the heat to be abstracted per hour by the air and water spray.

Air-Vapor Mixture Entering Condenser.

Pressure P_1 of the dry air:

$$P_1 = 29.0 - 0.7 \times 1.0314 = 28.28 \text{ in.}$$

(1.0314 = pressure of saturated vapor at temperature t = 80. deg. fahr.)

Volume V_1 of 1 lb. of dry air plus its vapor content, equation (435):

$$V_1 = \frac{0.754\,(459.6 + 80)}{28.28} = 14.41 \text{ cu. ft.}$$

Weight w_1 of vapor in 1 lb. of dry air:

$$w_1 = 0.7 \times 14.41 \times 0.00158 = 0.0159 \text{ lb.}$$

(0.00158 = density of saturated vapor at t_1 = 80 deg. fahr.)

Heat content H_a of 1 lb. of dry air above 0 deg. fahr.:

$$H_a = C_{pa}t_1 = 0.2414 \times 80 = 19.32 \text{ B.t.u.}$$

Latent heat r_v of vapor content in 1 lb. of dry air:

$$r_v = 0.7\,(14.41 \times 0.00158 \times 1047.4) = 16.68 \text{ B.t.u.}$$

(1047.4 = latent heat of saturated vapor at temperature t = 80.)

Total heat H_1 of mixture in 1 lb. of dry air:

$$H_1 = 19.32 + 16.68 = 36.00 \text{ B.t.u.}$$

Air-Vapor Mixture Leaving Condenser.

Pressure P_2 of the dry air:

$$P_2 = (29.0 + 0.294) - 0.98 \times 3.444 = 25.92.$$

(0.294 = value in inches of mercury of 4 inches of water pressure.)

Volume V_2 of 1 lb. of dry air plus its vapor content:

$$V_2 = \frac{0.754\,(459.6 + 120)}{25.92} = 16.89 \text{ cu. ft.}$$

Weight w_2 of vapor in 1 lb. of dry air:

$$w_2 = 0.98 \times 16.89 \times 0.00492 = 0.08143.$$

Heat content H_a' of the dry air in 1 lb. of mixture:

$$H_a' = C_p t = 0.2416 \times 120 = 29.00 \text{ B.t.u.}$$

Latent heat r_v' of vapor content in 1 lb. of dry air:

$$r_v' = 0.98\ (16.89 \times 0.00492 \times 1025.6) = 83.53 \text{ B.t.u.}$$

Total heat H_2 of the mixture in 1 lb. of dry air:

$$H_2 = 29.00 + 83.53 = 112.53 \text{ B.t.u.}$$

Heat taken up by 1 lb. of air plus water vapor in passing through the condenser

$$= H_2 - H_1 = 112.53 - 36.00 = 76.53 \text{ B.t.u.}$$

Total weight of dry air passing through condenser

$$= \frac{1{,}026{,}000}{76.53} = 13{,}400 \text{ lb. per hour.}$$

Total volume of air vapor entering the condenser

$$= 13{,}400 \times 14.41 = 192{,}960 \text{ cu. ft.}$$

Water absorbed per lb. of dry air

$$= w_2 - w_1 = 0.08143 - 0.0159 = 0.06553 \text{ lb.}$$

Total moisture absorbed or weight of spray to be injected

$$= 13{,}400 \times 0.06553 = 878.0 \text{ lb. per hr.}$$

For purpose of design it is sufficiently accurate to disregard the actual barometric pressure and assume it to be 29.92 in. With this assumption the problem may be readily solved by means of Table 138 or the curves in Figs. 713–4.

From Fig. 713 (for $t_1 = 80$ and $h_1 = 0.70$):

Wet bulb = 72.2, Dew point = 69.0.

$w_1 = 107$ grains $= 0.0153$ lb.

$H_1 = 35.5$ B.t.u.

From Fig. 714 (for $t_2 = 120$ and $h_2 = 0.98$):

Wet bulb = 119.4, Dew point = 119.2.

$w_2 = 555$ grains $= 0.0793$ lb.

$H_2 = 111$ B.t.u.

Moisture absorbed per lb. of dry air, and its vapor content,

$$w_2 - w_1 = 0.0793 - 0.0153 = 0.064 \text{ lb.}$$

Heat absorbed per lb. of dry air, and its vapor content,

$$H_2 - H_1 = 111 - 35.5 = 75.5 \text{ B.t.u.}$$

Since the moisture content per lb. of dry air at dew point is the same as that for all conditions of wet- and dry-bulb temperatures having that dew-point temperature,

Q.

R.

S.

U.